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USAAVLABS TECHNICAL REPORT 67-13

ADVANCEMENT OF SMALL GAS TURBINE ENGINE ACCESSORY TECHNOLOGY

By

Harry L. Richardson
Rocco M. Tommasini

April 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-297(T)
CURTISS-WRIGHT CORPORATION
WOOD-RIDGE, NEW JERSEY

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This Command has reviewed this report and concurs in the conclusions contained herein. The findings and recommendations of this report will be used to direct further efforts toward advancement of the state of the art of gas turbine engine accessories.

Task 1M121401D14416

Contract DA 44-177-AMC-297(T)

USAAVLABS Technical Report 67-13

April 1967

ADVANCEMENT OF SMALL GAS TURBINE ENGINE ACCESSORY TECHNOLOGY

Final Report

WAD R-431 F

by

Harry L. Richardson

Rocco M. Tommasini

Prepared by

Wright Aeronautical Division
Curtiss-Wright Corporation
Wood-Ridge, New Jersey

for

U.S. ARMY AVIATION MATERIEL LABORATORIES

FORT EUSTIS, VIRGINIA

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FOREWORD

The work reported herein is in fulfillment of Contract DA 44-177-AMC-297(T), which was conducted for the U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia. This report covers the study and evaluation of advanced concepts for small (2-to-5-pound-per-second airflow size) gas turbine engine accessories and controls. The study contract was conducted over a 12-month period between June 1965 and May 1966. It also includes the work performed by Honeywell, Aeronautical Division, under subcontract to Curtiss-Wright Corporation, Wright Aeronautical Division.

SUMMARY

This report describes the results of a study program conducted by the Wright Aeronautical Division for the U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia. The primary objective of the program was to investigate the feasibility of advancing the state of the art of small engine controls and accessories with a resultant decrease in weight, volume and increased system reliability. The specific program goals were as follows:

1. Determine the feasibility of eliminating all engine mounted accessory drive train reduction gears for 2-to-5-pound-per-second gas turbine engines.

2. Determine the feasibility of integrating various accessories into the basic gas generator.

3. Investigate and define advanced methods of energy extraction and addition.

4. Define a common control system for engines of the 2-to-5-pound-per-second airflow size.

5. Define an advanced accessory system for incorporation and test on a 1968-69 time frame demonstrator (2-to-5-pound-per-second airflow) engine.

The program was divided into three phases of problem definition, accessory design coordination and engine/system integration. The parametric engine characteristics defining the controls and accessories requirements utilized an advanced engine design reflecting both the 2-to-5-pound-per-second airflow size and advanced engine components presently under study by USAAVLABS. The design coordination phase was conducted by invitation to representatives from all the leading accessory manufacturers to participate and contribute conceptual designs and ideas directed toward the advancement of technology in the small gas turbine controls and accessory field. The accessory manufacturers were encouraged to combine, integrate dual-function elements, integrate components into basic engine hardware and if necessary deviate from normal practice for engine incorporation of new designs, concepts and ideas. This type of approach necessitated the utilization of a "Problem Statement" type specification instead of the usual component specification in an effort to provide the accessory manufacturer with a complete engine requirement and to provide the component designer a high degree of design latitude.

The engine/system integration utilized the results of advanced engine work being conducted by the Wright Aeronautical Division to define a realistic 1969 time frame gas generator design. This gas generator design was used for the purpose of investigating two basic controls and accessories installation concepts. The first was a miniaturized high speed version of a conventional accessory gearbox and the second featured the integration of the rotating accessories into a nose mounted gearbox.

Honeywell, Aeronautical Division, having done a significant amount of work on fluidic devices and their application to advanced turbine engine control systems, was retained to conduct a controls system analysis and provide a conceptual design of a fluidic engine control system with fluidic sensors applicable to gas generators in the 2-to-5-pound-per-second size. With the exception of Honeywell, all the accessory manufacturers were asked to voluntarily contribute their conceptual designs and ideas. The conceptual nature of the program and the lack of direct support resulted in a good percentage of the accessory manufacturers electing not to participate. For this reason, some of the conclusions cannot be supported completely by accessory manufacturers' inputs. However, opinions and trends were obtained in many cases to justify the basic conclusions.

The overall program resulted in uncovering areas where substantial gains could be achieved in decreasing the installed volume and weight. The following conclusions were reached:

1. The accessory gearbox cannot be eliminated in total. However, the integration of the shaft-driven accessories into the accessory gearbox utilizing high density packaging and maximum component speed makes for a substantial gain in the areas of volume and weight.

2. Within limits, accessory drive speeds can be increased over current operating speeds by at least a factor of two.

3. Module packaging can utilize engine and/or aircraft space more economically and can effectively place vulnerable components in more favorable environments.

4. The dual-function component concept should be studied in greater detail.

5. Engine starting can best be accomplished by using a hydraulic starting system.

It was also concluded that fluidic technology can be applied satisfactorily to the control of small gas turbine engines. However, much additional development work is required in order to obtain the full benefits of this new concept.

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ILLUSTRATIONS (Cont'd)

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TABLES

SYMBOLS

W_a	airflow rate
W_f	fuel flow rate
N_1	rotational speed, gas generator spool
N_d	rotational speed, gas generator spool, design
ΔP	differential pressure
P	absolute pressure
I_P	polar moment of inertia
T	temperature
SFC	specific fuel consumption, W_f/SHP
SHP	shaft horsepower
Q	torque
U	mean wheel speed
h	specific enthalpy
P_r	pressure ratio
P_c	control reference pressure
P_a	ambient pressure
T_{2_T}	compressor inlet total temperature
c	acoustic velocity
g	acceleration due to gravity
K	adiabatic exponent
R	gas constant
TIT	turbine inlet temperature
η_{ax}	axial adiabatic efficiency
δ	relative absolute pressure, P/P_o
θ	relative absolute temperature, T/T_o

SYMBOLS (CONT)

P_{3_T}	compressor discharge total pressure
P_{2_T}	compressor inlet total pressure
T_B	combustor time constant
T_{4_T}	turbine inlet total temperature

INTRODUCTION

In the propulsion field, progress has been made to a point where thrust/weight ratios of 20 to 22 are presently being achieved on experimental lift engines. These tremendous strides in the last 20 years in the field of gas turbine engines have advanced the technology to a degree where VTOL aircraft have become a reality. However, in the area of engine mounted accessories, the state of the art has for all practical purposes remained static. This lack of advancement in the accessory field is clearly evident in the new generation of lift engines and in the small (2-to-5-pound-per-second airflow) gas generators presently under investigation by the U. S. Army Aviation Materiel Laboratories (USAAVLABS), Fort Eustis, Virginia.

The current USAAVLABS component programs are intended to reduce the specific weight of the 2-to-5-pound-per-second gas generators by approximately 50 percent, with corresponding decreases in specific fuel consumption (SFC) and installed volume. If current state-of-the-art accessories are installed on these advanced gas generators, the weight of the accessory system will exceed the gas generator weight for the 2-pound-per-second machine. Therefore, the primary objective of the program is advancement in the state of the art of small engine accessory systems with resultant decreases in weight, installed volume, and unit cost and with increased reliability. The secondary objective is the total elimination, if possible, of the accessory drive train reduction gears which account for a large portion of the small gas generator weight. The overall program goals are outlined as follows:

1. Determine the feasibility of eliminating all engine mounted accessory drive train reduction gears for 2-to-5-pound-per-second gas generators.

2. Determine the feasibility of integrating various accessories into the basic gas generator.

3. Investigate and define advance methods of energy extraction and addition.

4. Define a common control system for engines of the 2-to-5-pound-per-second airflow size.

5. Define an advanced accessory system for incorporation and test on a 1968-69 time frame demonstrator (2-to-5-pound-per-second) engine.

It is evident from the program goals that the main purpose of the program was to uncover new methods, concepts and ideas directed toward the advancement of controls and accessories as related to the small gas generator. This study effort could lend impetus and direction to future USAAVLABS research and development efforts on controls and accessories for advanced engines.

The study program included the following control and accessory areas:

1. Fluid pumps
2. Generators and alternators
3. Starters
4. Speed sensing devices
5. Ignition systems
6. Temperature indicating systems
7. Variable geometry devices
8. Engine controls

In order to provide the accessory manufacturers with component design requirements, a "Problem Statement" type specification was agreed upon as being the least restrictive and the most informative from a systems standpoint; it also provided the accessory designer a high degree of design latitude with minimal boundary restrictions. The Problem Statement is included with this report as Appendix I.

To provide realistic characteristics for small gas turbine engines, for engine control studies, a Wright Aeronautical Division advanced engine design was utilized. The engine design characteristics are as follows:

W_a = 4.483 lb/sec

TIT = 2000°F

N = 50,000 rpm

HP = 721

SFC = .366

P_r = 8.0:1

I_p = 0.14 in-lb-sec^2

The characteristics of this design were then extrapolated as required to provide a family of engines within the 2-to-5-pound-per-second airflow size. The engine characteristics developed in this manner were then coordinated with USAAVLABS to incorporate and reflect the variations imposed by the designs of other engine manufacturers. Therefore, while the characteristics shown in the Problem Statement are not strictly those of specific engines, it reflects, in terms of controls and accessories, the requirements of a projected advanced gas turbine engine of the type under consideration for the 1968-69 time frame demonstrator engine.

The Problem Statement was coordinated with USAAVLABS and approval was obtained to distribute this document to the accessory manufacturers that had been selected to participate in the program. When the Problem Statement was distributed, the vendors were invited to attend a briefing at the Wright Aeronautical Division on July 21, 1965. This briefing was attended by 25 individuals representing 19 controls and accessory manufacturers. USAAVLABS personnel were also present at this meeting. A complete list of those companies represented at the meeting is presented in Appendix II.

Many vendors' representatives indicated an interest in the concept of proceeding with advanced controls and accessories at this time to insure their availability for future engine requirements. However, many vendors subsequently elected not to participate in the programs for various reasons, while others were able to make only token contributions.

A subcontract was negotiated with the Aeronautical Division of Honeywell to cover a study of the application of fluidic technology to the control of small gas turbine engines. The Honeywell final report is presented as Appendix III of this report. The significant results of this program are presented in the following sections of this report.

ENGINE ACCESSORIES

FLUID PUMPS

When considering the fluid pump requirements for small gas turbine engines, it becomes immediately evident that size and weight are closely related to pump speed. It would appear, therefore, that a two-fold size and weight advantage could be obtained by running the fluid pumps at engine rotor speed. The fluid pumps would be much smaller, and the usual gearing would be eliminated. This concept was advanced in the Problem Statement of Appendix I. The fluid pump requirements were outlined in detail in the Problem Statement but can be generalized as indicated below.

Component	Flow Requirements (gpm)	Discharge Press. (psi)	Maximum Speed (rpm)
Fuel pump	0.4 to 1.0	600	40,000 - 100,000
Hydraulic pump	2.5 to 6.0	3000	40,000 - 100,000
Lube pressure pump	1	70	40,000 - 100,000
Lube scavenge pump	2	30	40,000 - 100,000

There were no vendor responses that completely reflected the requirements of the Problem Statement. One manufacturer* did provide general information on a series of small, high speed pumps that they have developed for various missile applications. These pumps are of the gear type and cover a flow range up to 23.5 gpm in three basic sizes. As indicated in Figure 1, the smaller of the three sizes is a cartridge that is 1.312 inches in diameter and 1.130 inches long. This unit can operate at speeds up to 60,000 rpm and will provide pressures up to 3000 psi. This unit is actually oversized for the 2-to-5-pound-per-second engines. However, the cartridge configuration and high speed is directly applicable. Since these pumps have been developed for missile applications, the design life of such a unit is probably measured in minutes. It would be of interest to evaluate a pump of this type from a durability standpoint to determine if adequate life can be attained. At the same time it would be appropriate to run contaminated fuel tests to obtain a realistic picture of this aspect of pump performance.

The pump manufacturer cited above has indicated that considerable attention has been given to the cavitation problem associated with high speed. The inlet port is large, allowing fluid to enter the tooth spacing both radially and axially for almost one quadrant of each gear.

Reference to Figure 1 shows that the smaller of the three pumps will deliver adequate flow to meet the requirements of the fuel and lubrication systems while running at speeds between 15,000 and 20,000 rpm. In order to meet the hydraulic system requirements, a slightly larger pump is required if the speed is held at this level. It is suggested that there is little

* The Adel Division of General Metals Corporation

1. Performance shown at ΔP = constant = 3000 psi.
2. Curves do not reflect temperature, altitude or density variations.
3. Modification can be made to pumps to meet flow, pressure and speed requirements not covered by curves.

Basic Cartridge Dimensions

Basic Size Number	10	20	30
Gear Size, in.	5/8	15/16	1 1/4
Cartridge Dia., in.	1.312	1.967	2.622
Nominal Length, in.	1.130	1.732	2.318
Nominal Weight, lb	0.31	0.93	2.31

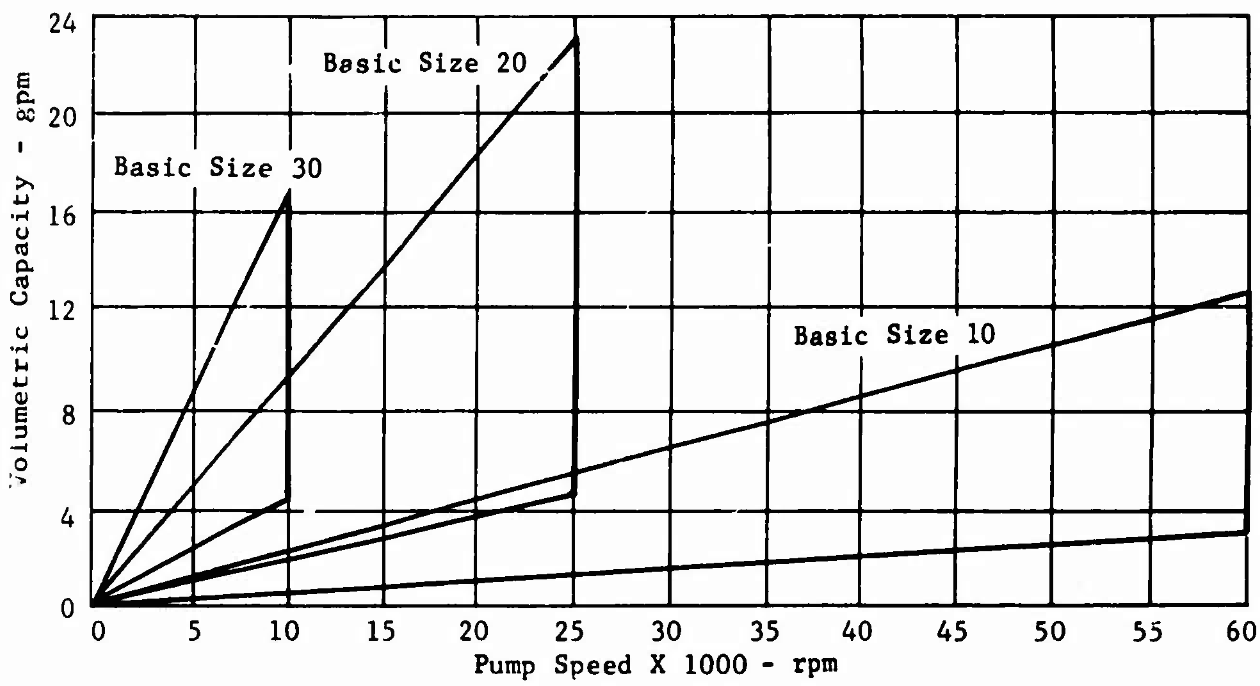


Figure 1. Adel Division, General Metals Corp., High Speed Gear Pump Characteristics.

to be gained in terms of size and weight by running the pumps at speeds above 20,000 rpm. Also, by holding the speed at 20,000 rpm, problems such as excessive wear and cavitation will be minimized. In addition, the manufacturing costs of very small, ultra high speed pumps would tend to be high due to the order of precision involved. By making this compromise in speed, it is expected that fluid pumps suitable for use on small gas turbine engines could be made available by a reasonable development effort.

The above philosophy might also be applied to the miniature vane type pumps offered for missile application by a second manufacturer.* A typical 1.0-gpm cartridge is approximately 1.125 inches in diameter and 0.75 inch long. Current units are designed to operate at pressures up to 1000 psi and speeds up to 10,000 rpm. By the selection of proper materials it should be possible to obtain an engine rated version of this unit capable of operating at speeds up to 15,000 rpm.

Centrifugal pumps are apparently not applicable to the requirements of small engines. The representatives of one pump manufacturer** have stated that such units cannot be scaled to the size required without a tremendous sacrifice in efficiency. This comment was also received from a second accessory manufacturer*** relative to the use of ultra high speed centrifugal pumps in a constant speed drive device.

New and unique pump concepts were not revealed by this program; therefore, it is recommended, for the time period of interest, that any anticipated development effort be expended in the direction of extending the durability of gear and vane type pumps. The design operating speed should be in the 15,000-to-20,000-rpm range, and the overall configuration should be of the cartridge type.

ALTERNATORS AND GENERATORS

As defined in the Problem Statement of Appendix I, an objective of this program was to investigate the feasibility of extracting electrical power from the gas generator for use in the various vehicle subsystems. The amount of electrical power to be extracted was established as 1 percent of rated engine power and this amount of power was to be available at idle speed. For the 2-pound-per-second engine this would be 3.0 to 6.0 horsepower (depending upon the engine pressure ratio) at engine speeds between 35,000 and 40,000 rpm. The corresponding values for the 5-pound-per-second engine were 8.0 to 16.0 horsepower at speeds of 14,000 to 16,000 rpm.

A look at conventional electrical aircraft type equipment points up some of the problems associated with the extraction of electrical power from small gas turbine engines. For instance, a 30-volt,100-ampere (5.7 horsepower) starter/generator is 8 inches long and 5 inches in diameter and

* Vickers, Incorporated
** Thompson Ramo Wooldridge, Incorporated
*** Sundstrand Aviation

weighs 17 pounds. A similar unit capable of developing 12 horsepower is 11.5 inches long and 6.5 inches in diameter. This unit weighs 46 pounds. A 10-kva, 115-volt,400-cps brushless generator is 12.5 inches long and 6.5 inches in diameter and weighs 30 pounds. Units of the type described here are designed to operate at speeds ranging from 4000 rpm to 12,000 rpm and as such would require a speed reducing gear train to adapt them to conventional small gas turbine engines.

It is evident that, from a weight and volume standpoint, conventional equipment is completely unacceptable relative to the objectives of this program; however, two possibilities exist which could improve this situation. The first, a very extensive advancement in the state of the art of rotating electrical machinery, would be necessary for these devices to operate at much higher speeds. It is anticipated that a significant reduction in size would result from high speed operation.

The second possibility is to reduce or eliminate the electrical secondary power requirement for the small gas generator. This concept is discussed in detail in the section of this report covering starters. It appears that many benefits could be obtained by making the gas generator self-sufficient and depending upon secondary electrical power being taken from the drive train. This approach would permit the development of a standardized gas generator designed to take full advantage of miniaturization and advanced packaging techniques. In essence, it has been concluded that the attainment of a self-contained gas generator unit of small volume is not likely if a substantial amount of electrical secondary power is required. However, as noted in the discussion of starters, it is possible to extract a significant amount of secondary power hydraulically. Therefore, if the vehicle requires electrical power while the gas generator is at idle, it would be possible to make use of a remote, hydraulic motor driven generator.

In the course of investigating the availability of high speed electrical equipment, two possible sources were revealed. One manufacturer* provided information covering two high speed generators. The smaller of the two is a 5-ampere, 24-volt device that was designed for a shaft speed of 52,500 rpm. This company feels that the speed of this unit might be extended to 100,000 rpm. The generator is of the permanent magnet type and is 2 inches in diameter and 1.5 inches long. This unit does not include bearings, and the rotor is designed to mount on a shaft integral with the drive unit. A larger unit is of the inductor electromagnetic type and develops 15 amperes at 24 volts at a speed of 40,000 rpm. This unit is 3.5 inches in diameter and 2 inches long.

The above units were designed primarily for battery charging but can also be used to provide a speed signal. The permanent magnet design provides a voltage and/or frequency signal that is directly proportional to rotor speed. The inductor electromagnetic design requires a constant field voltage in order to provide a proportional voltage and/or frequency signal.

* The R. E. Phelon Company

When asked if the basic concepts could be scaled to provide an increased electrical output, the manufacturer estimated that an 8-horsepower machine operating at 40,000 rpm would be cylindrical in shape and would be about 7 inches in diameter and 3 inches long. The unit would weigh approximately 20 pounds. However, concern was expressed that the armature stresses resulting from the high operating speeds would be excessive to the point that the design would not be feasible from a mechanical standpoint.

A second vendor* provided information on small high frequency generators that might be suitable for special applications in which wild or random frequency is acceptable. One unit that produces a 600-watt output at 60,000 rpm, shown in Figure 2, is used in a missile application and is driven by an integral hot gas turbine that is approximately 1.5 inches in diameter. The unit weighs 8.5 ounces. The variable speed-constant frequency (VSCF) concept as a means of providing a constant output frequency from their devices is not used in this design.

Discussions were held with a third vendor** relative to the application of their variable speed-constant frequency approach to the electrical requirements of small gas turbine engines. In an application that required constant frequency electrical power, the use of VSCF equipment would eliminate the need for a constant speed drive (CSD). However, since the electronics package associated with VSCF does not scale well with size, there is a question as to the weight and volume trade-off between alternator/CSD and VSCF. It was indicated that they do not have development programs under way that are applicable to small gas turbine engine requirements. However, it was stated that if the design of such units were attempted, they would prefer that the bearings be integral with the engine.

From the comments received from the manufacturers of rotating electrical equipment, it appears that the objectives set forth in the problem statement were optimistic and that, in the near future, high speed electrical machines cannot be expected to provide the output desired at the rotor speeds specified. For this reason, it is recommended that additional consideration be given to the concept of zero or minimum electrical secondary power from the gas generator. In addition, it is suggested that a detailed study of specific engine applications might indicate that the high speed-low output electrical devices might be used to advantage.

STARTERS

One of the primary objectives of a starter or starting system for small gas turbine engines is self-sufficiency. This is particularly important from the standpoint of austere site operation. The starting system must be simple and maintenance free. It must also be of light weight and low volume. It is also desirable that the starter serve a useful purpose once the engine has been started. There is also a requirement that the starting system permit three starts or start attempts without having to resort

* Varo, Incorporated
** General Electric

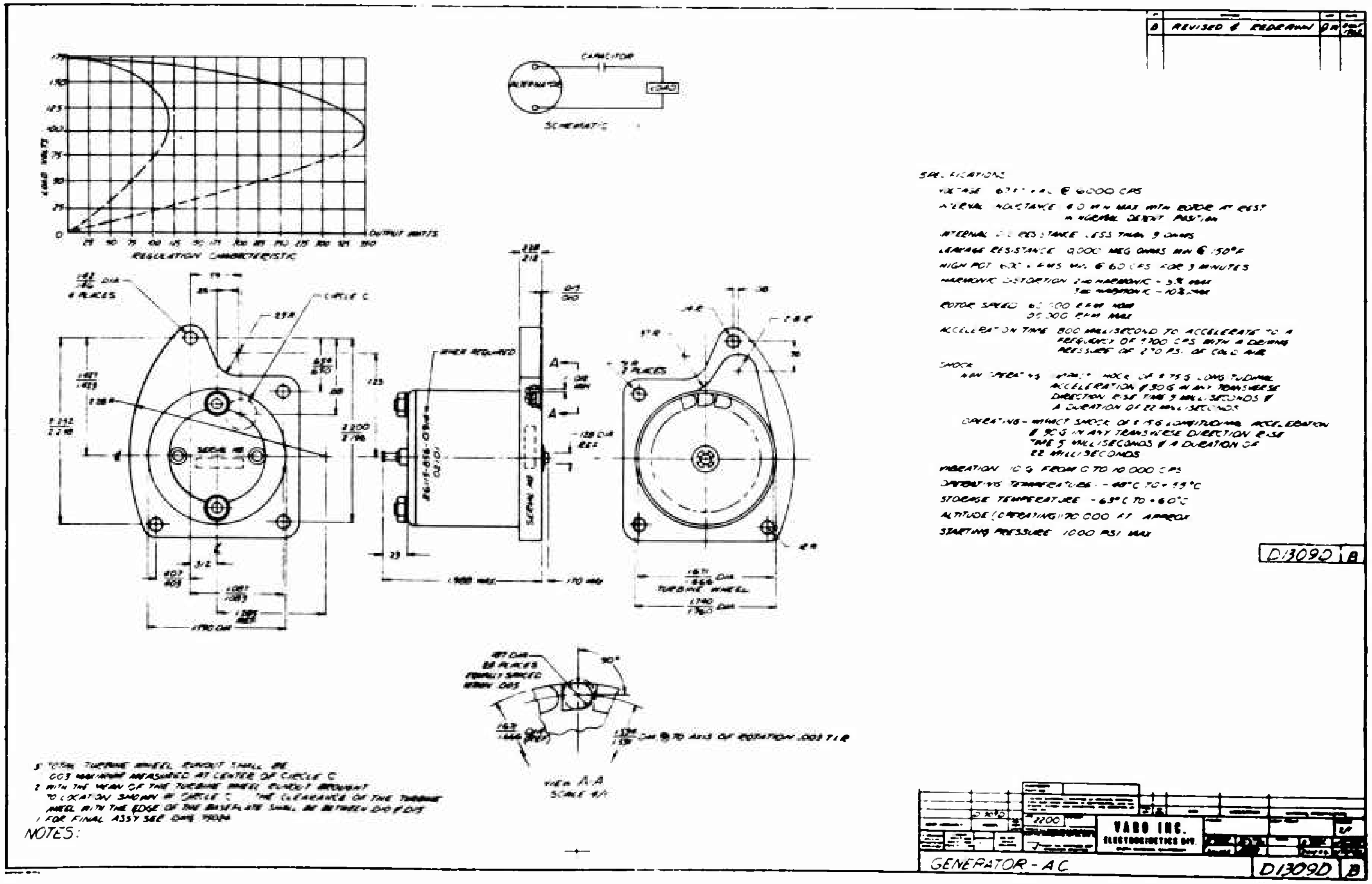


Figure 2. Varo, Inc., High Speed Alternator.

to auxiliary energy sources. As will be pointed out later, this last requirement is very difficult to satisfy completely.

In the course of this investigation, three primary types of starting systems were considered in some detail:

1. Electrical starter

2. Combustion starter

3. Hydraulic starter

Other approaches such as air turbine, cartridge, and impingement arrangements were considered briefly but were eliminated as candidate systems. The reasons for eliminating these systems were associated with complexity, the need for ground support equipment and logistics. Air turbine starters are, in themselves, relatively complex and bulky. The efficiency of such devices is reasonable, but a considerable quantity of pressurized air is required for operation. This requirement prevents self-sufficiency from being attained and complicates operation at remote sites. In addition, the air turbine does not meet the dual-purpose requirement. Once the engine has been started, it becomes dead weight. It is doubtful that the development of a scaled-down air turbine starter is warranted for small gas turbine engines.

Impingement starting is attractive from the standpoint of engine installed weight and volume. However, the inherent low efficiency of such a system would rule it out. The air supply requirements would probably exceed those of an air turbine by at least a factor of five or six. Here again self-sufficiency is impractical. There is a possibility of applying impingement starting to a multiengine application. In such a case the second engine could be started by impingement by bleeding the compressor of the first engine after it has been started.

Cartridge starting was eliminated as a candidate because of the cost aspects associated with such a system. It is assumed that the duty cycle of the small engines being considered would be such that starts are quite frequent. When the cost of starting is related to the per mission cost of engine operation, the result is not acceptable. Also there would be considerable expense associated with the development of the starter proper (a scaled-down turbomachine) and the cartridge. Logistics is also an important factor relative to cartridge starters.

Throughout the following discussion on starters, a 4-pound-per-second engine is used as a reference. This engine is also used for the installation study covered in a later section of this report.

Electrical Starter

The electrical starting system is attractive to some degree because of the relative simplicity of such a system. However, when weight and volume are

taken into consideration, the use of such a system becomes questionable. A development effort being conducted by one company* was directed toward the reduction of size and weight of this type equipment; however, their advanced electrical starter program was terminated during the course of this study program according to information received, and no conclusive information is available.

No vendor information was received during this study program that would significantly improve the state of the art in the area of electrical starters. In order to have a basis for comparison, an electrical starting system was sized for the 4-pound-per-second engine used in the installation study. The starter requirement, in terms of torque versus speed, is shown in Figure 3. As indicated, the starter cutoff speed occurs at an engine rotor speed of 21,000 rpm. In order to match the above cutoff speed it is necessary to make use of a speed increaser between the starter and the engine rotor.

The electrical starter torque characteristic of a typical state of the art electrical starter is also shown in Figure 3**. Starter torque is somewhat marginal in the vicinity of light-off but is considered satisfactory for the purposes of this study. The essential elements of the electrical starting system are shown in Figure 4 along with a weight summary. As indicated, the estimated weight of the complete system is 66 pounds. As will be shown later, the weight of an electrical starting system is not currently competitive with other starting arrangements.

In the various contacts with the manufacturers of rotating electrical equipment, it has become evident that significant breakthroughs are not anticipated in the near future. The centrifugal stresses associated with high rotating speeds that might yield lower weight and smaller size have been cited as one reason for the lack of progress. It must be concluded on the basis of information obtained during this study that electrical starting systems are not compatible with the overall objectives of USAAVLABS.

Combustion Starter

Information was submitted to this contractor concerning the application of an advanced starter concept to small gas turbine engines. Complete details on the principles of operation of this starter are not available. However, it is known to be a liquid fueled-air breathing device. It is also known that a unit of this type is under development for U. S. Army Engineering Research and Development Laboratories (USAERDL).

The company developing this equipment*** indicated that their starter concept was compatible with the direct drive requirement in the 5-pound-per-second engine size but that gearing would be required if this starter were

* Pratt and Whitney Aircraft Division of United Aircraft Corporation
** Courtesy of the Bendix Redbank Division
*** The Rocketdyne Division of North American Aviation Corporation

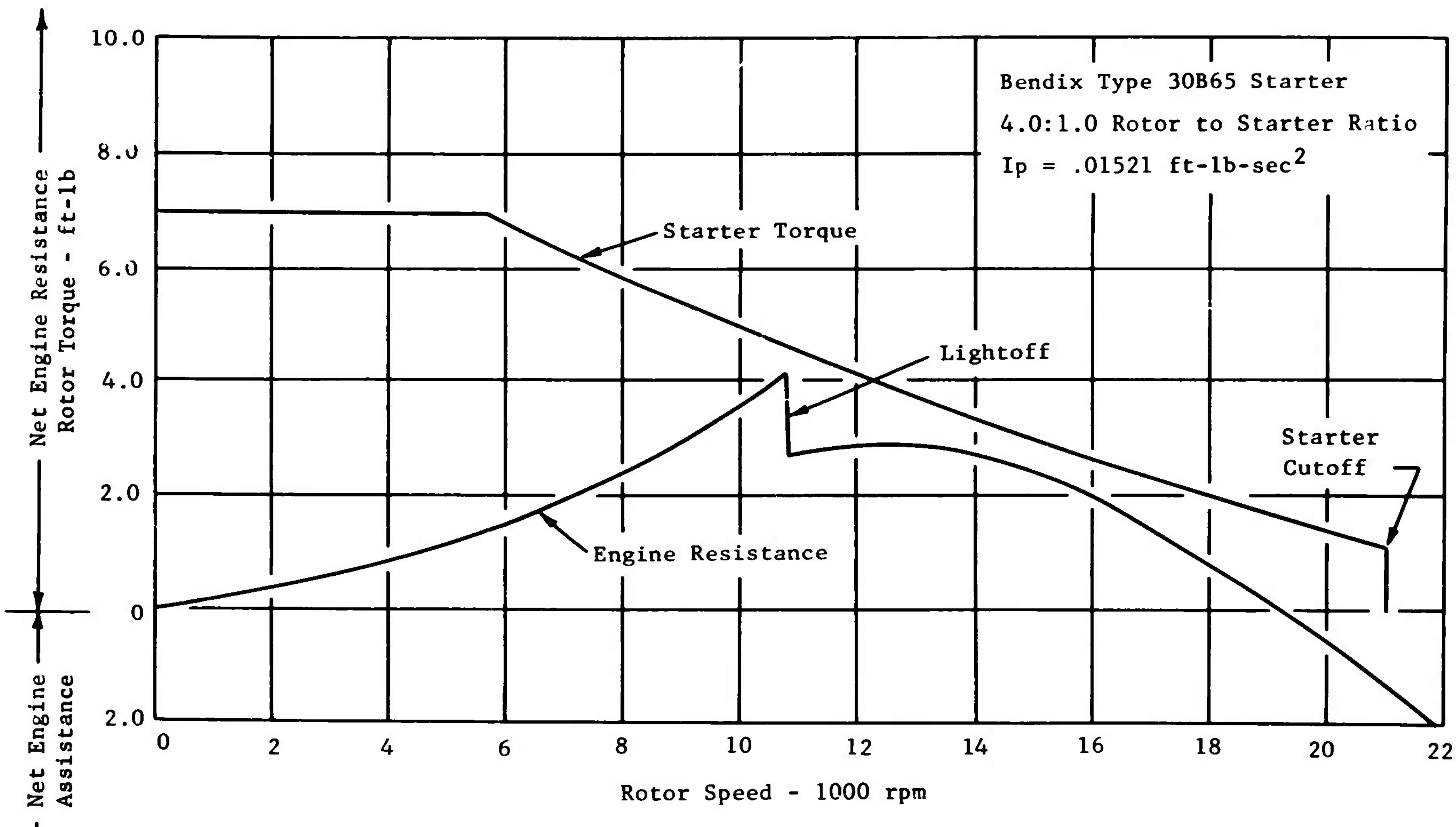
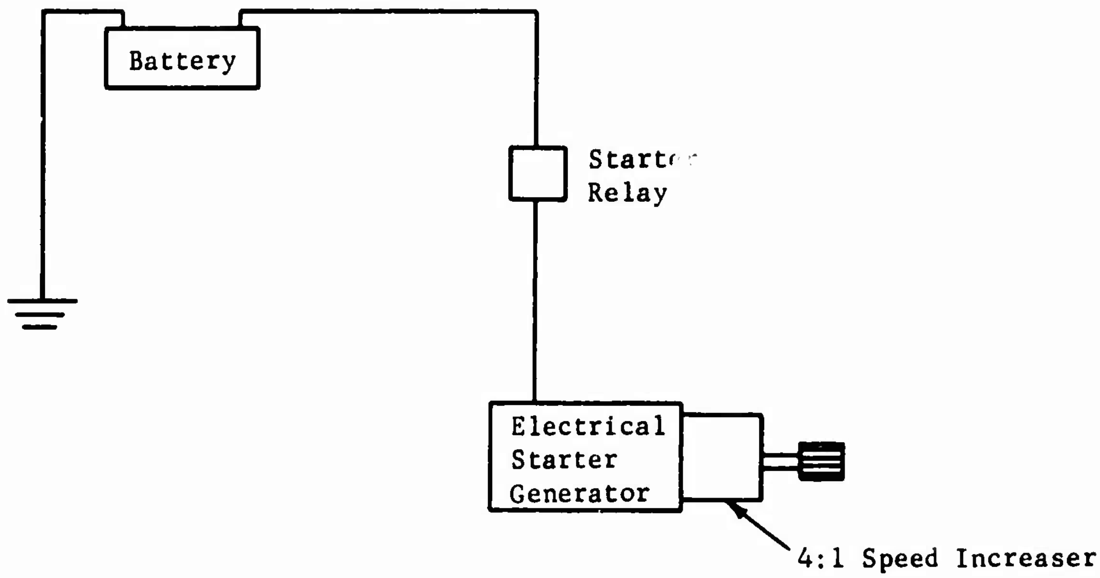


Figure 3. Electrical Starter Characteristics.



Weight Summary:

Starter	26.0 lb
Battery	30.0
Relay	5.0
Wiring	2.0
Speed Increaser	3.0
	66.0 lb

Figure 4. Electrical Starter System Schematic.

used on smaller engines. Improvement in this area of higher speed is expected as technology is extended.

The estimated characteristics and dimensions of the combustion starter are presented in Table I, as provided by the manufacturer, for both the 5- and 2-pound-per-second engines. The information contained in this chart pertains to current technology. The outline dimensions of a unit currently under development are shown in Figure 5. This package is not quite large enough for the 5-pound-per-second engine but is larger than would be required for the 2-pound-per-second engine. The particular shape of the package is based on a specific application and could be re-arranged to some degree if required.

Table II presents the manufacturer's estimate of the reduction in size and weight that might be attained as a result of additional development effort. From this information it can be determined that a combustion starter for the 4-pound-per-second engine would weigh approximately 23 pounds at the completion of an extensive development program. As noted in Figure 3, which relates to electrical starters, a current electrical unit suitable for the 4-pound-per-second engine weighs 26 pounds.

The characteristics of the combustion starter when applied to the 4-pound-per-second engine are shown in Figure 6. The starter torque curve is derived from data supplied by the manufacturer. The curve indicates adequate torque throughout the speed range.

Relative to the objectives of this program, the combustion starter approach is not truly self-sufficient, since it requires fuel and air under pressure and electrical energy for operation. The use of fuel and air stored under pressure has been suggested and is considered to be generally satisfactory. However, manually operated fuel and air pumps will be required to replenish the supply in the event of an excessive number of false starts. Such an arrangement, which provides a means for recharging the fuel and air accumulators manually, is shown in Figure 7. In normal operation the accumulators are recharged as soon as the engine starts. It is assumed that the air will be stored at about 145 psia. This pressure level would permit a 16:1 compressor to recharge the storage tank at something less than full engine power. At 145-psia storage pressure, the tank would be about 7.0 inches in diameter to accommodate 170 cubic inches of air for one start attempt. To obtain three starts from the storage tank, the volume would be 510 cubic inches and the sphere diameter would be 10 inches. The respective weights of the above storage tanks would be approximately 2.5 pounds and 7.5 pounds. It is estimated that the corresponding weights of the fuel storage tanks would be 1.0 pound and 1.75 pounds. The estimated weights of the combustion starter system for both a single-start and a three-start configuration are summarized in Figure 7. As noted, the single-start system weighs 35.0 pounds and the three-start system weighs 40.75 pounds. Battery weight is not charged against the starter system, since the power requirement is small and electrical power such as this might otherwise be available.

TABLE I

EXISTING ROCKETDYNE COMBUSTION STARTER PARAMETERS

	5 lb/sec Engine	2 lb/sec Engine
Weight, lb	40	24
Volume, ft^3	1.32	0.81
Dimensions (approximate), in.	9 x 13 x 19.5	8 x 11.5 x 15.25
Stall Torque (sea level, +60°F), lb-ft	14	3.1 (with gears)
Peak Horsepower (sea level, +60°F)	15.5	9.0
Rated Fuel Flow (sea level, +60°F), lb/hr	128	76
Fuel Types	Gasoline, Compression-Ignition, JP-3, JP-4, JP-5	

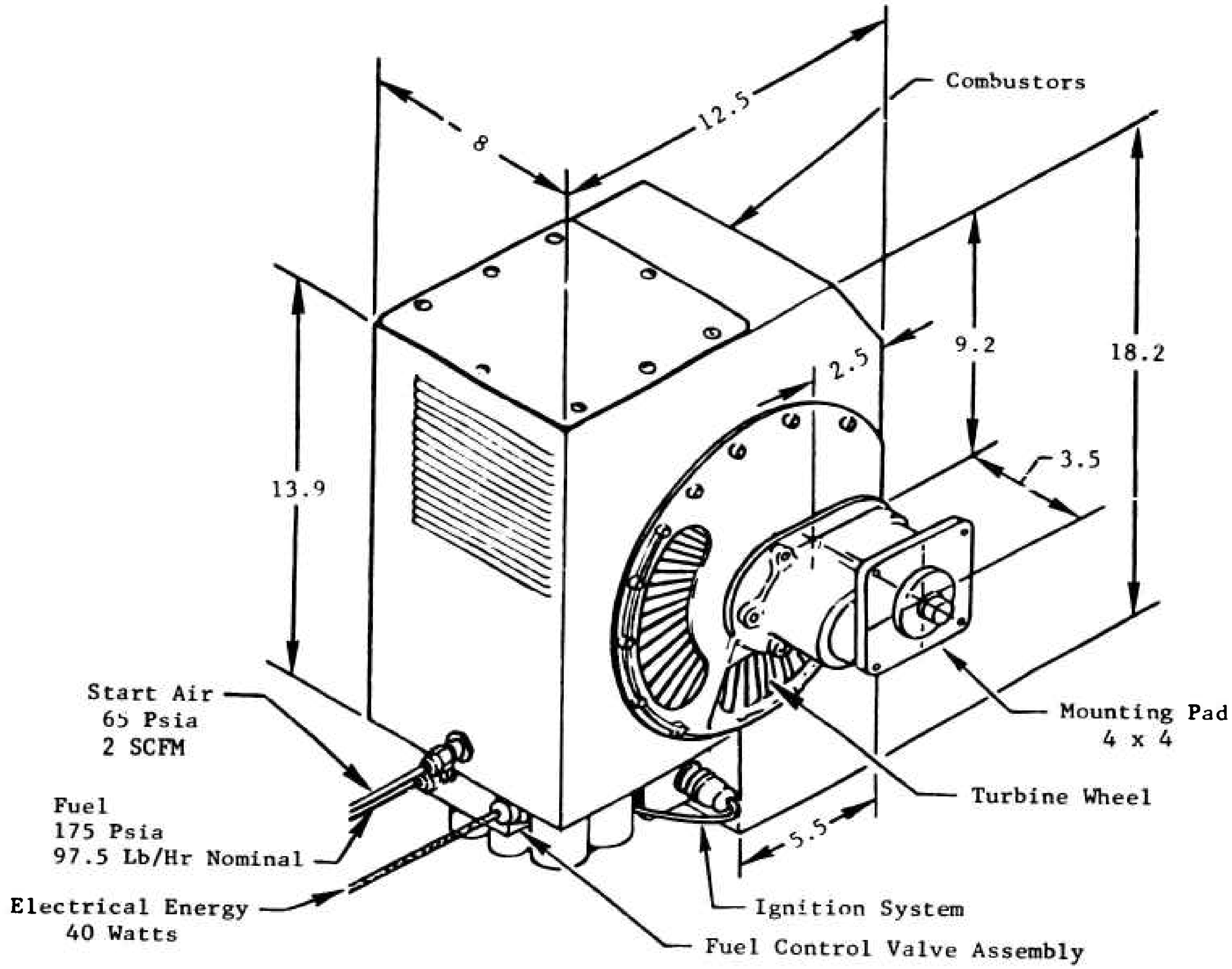


Figure 5. Rocketdyne Combustion Starter Dimensions.

TABLE II

ROCKETDYNE COMBUSTION STARTER DESIGN

	5 lb/sec Engine		2 lb/sec Engine	
Development Status*	A	B	A	B
Weight, lb	32.0	26.0	19.0	16.0
Volume, ft^3	0.79	0.53	0.49	0.33
Overall Dimensions, in.	10.0D x 17.5L	9.0D x 14.5L	8.7D x 14.2L	7.5D x 13L
Rated Fuel Flow, lb/hr	128.0	96.0	76.0	57.0

*Development Status A - Refined packaging application and some improvement of precut technology

Development Status B - Improvements in several performance and packaging characteristics

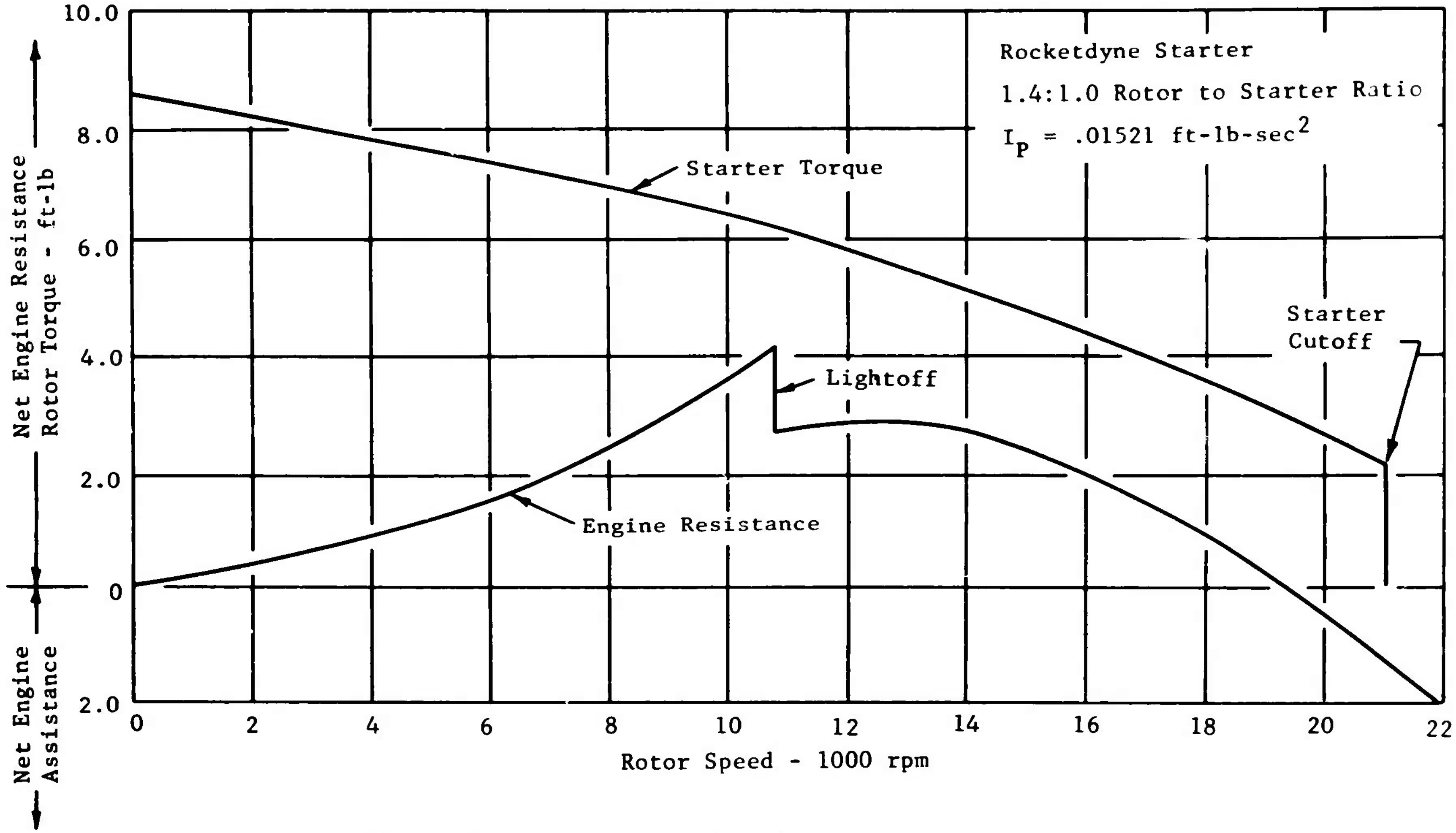
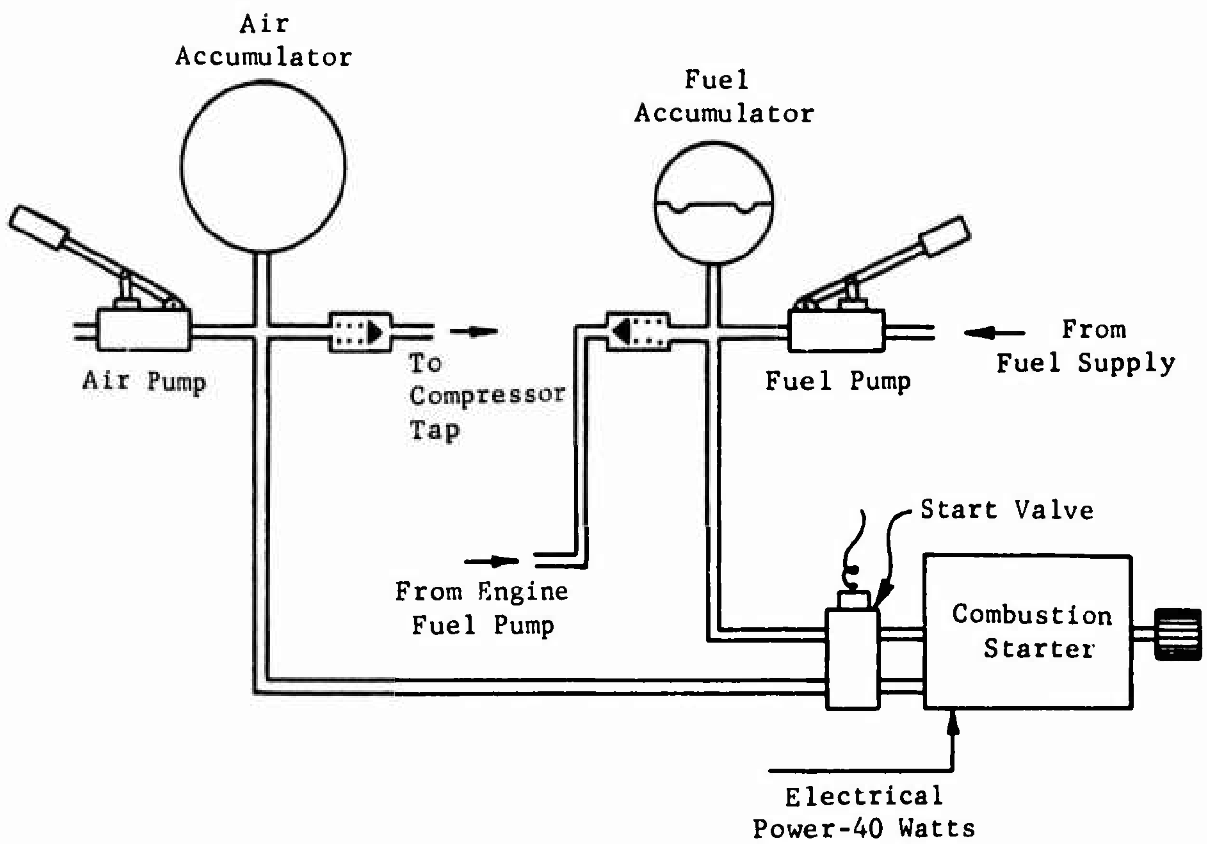


Figure 6. Rocketdyne Combustion Starter Characteristics.



WEIGHT SUMMARY

	Single Start	Three Starts
Starter	23.0 lb	23.0 lb
Air Accumulator	2.5 lb	7.5 lb
Fuel Accumulator	1.0 lb	1.75 lb
Air Pump	3.0 lb	3.0 lb
Fuel Pump	2.0 lb	2.0 lb
Valving	2.0 lb	2.0 lb
Lines and Misc. Hdw.	1.5 lb	1.5 lb
Total	35.0 lb	40.75 lb

Figure 7. Combustion Starter System Schematic.

Also, it should be noted that no weight is included for the 1.4:1 speed increaser indicated on Figure 7. It is assumed that the starter could be properly designed to crank the engine at rotor speed. The 1.4:1 ratio was used to match the available starter data to the requirements of the 4-pound-per-second engine.

Another disadvantage of the combustion starter concept is that it cannot be arranged to provide a dual function; i.e., it serves no useful purpose once the engine has been started.

While this combustion starter concept embodies some rather attractive features such as cold-start capability and partial self-sufficiency, it does not lend itself to the objectives of this program. The weight of the combustion starter system is excessive and the volume of the engine mounted portion of the system is completely at odds with the requirement for compactness. However, this concept might be used to advantage in applications in which weight and volume are not critical.

In addition, it is suggested that this starter concept might be used in an accessory drive package or auxiliary power unit. It is assumed that the unit can be designed for continuous duty and packaged with the necessary gear reduction to drive electrical power generating devices and hydraulic pumps.

Hydraulic Starter

A report was submitted by a well-known accessory manufacturer* covering a starting system termed "Fuel-Draulic" which described a system considerably oversized for the class of engines under consideration. However, the pertinent features of this system are of interest and are presented for the record. The system is shown schematically in Figure 8. A two-stage centrifugal pump is driven by a self-contained gas turbine type auxiliary power unit (APU). This pump operates on engine fuel and develops a pressure of 1500 psi. The pump discharge can be directed by way of a selector valve either to a liquid turbine that is used as engine starter or to a similar liquid turbine that is used to power the aircraft accessory drive gearbox. The fluid turbine is of the axial flow type.

The system reported will start engines requiring 30-40 horsepower and is therefore not directly applicable, and it is questionable that such a system could be scaled to meet the requirements of small gas turbine engines and show a weight and volume advantage. In addition, the need for a prime mover for the fluid pump also implies a complexity that is undesirable.

To supplement vendor inputs regarding hydraulic starting, this contractor undertook a design study to investigate the merits of such a system. The resultant hydraulic starter system is shown in Figure 9. Hydraulic fluid is stored at an initial pressure or 4000 psi in a conventional accumulator.

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* Accessories Division of Thompson Ramo Wooldridge, Inc.

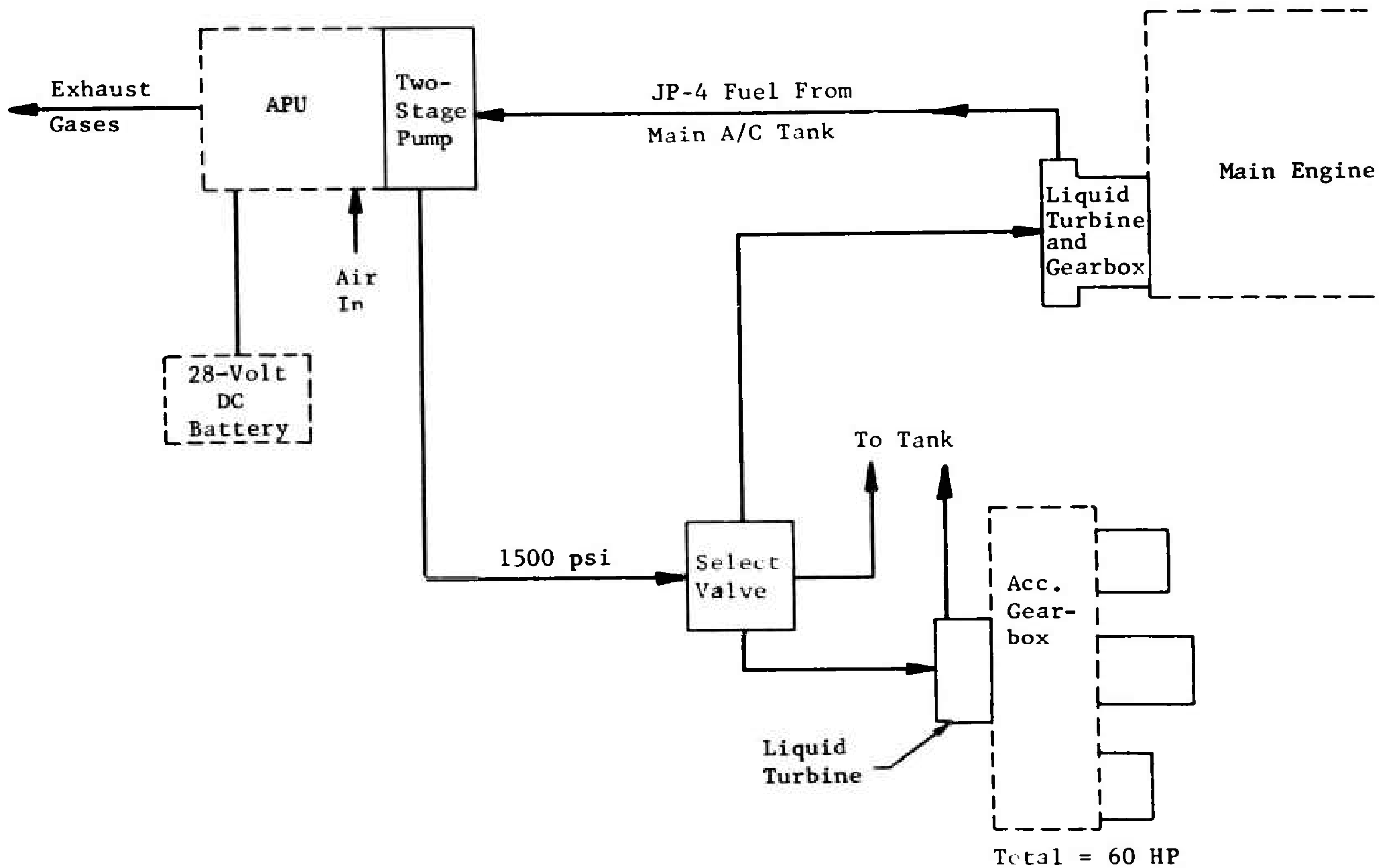
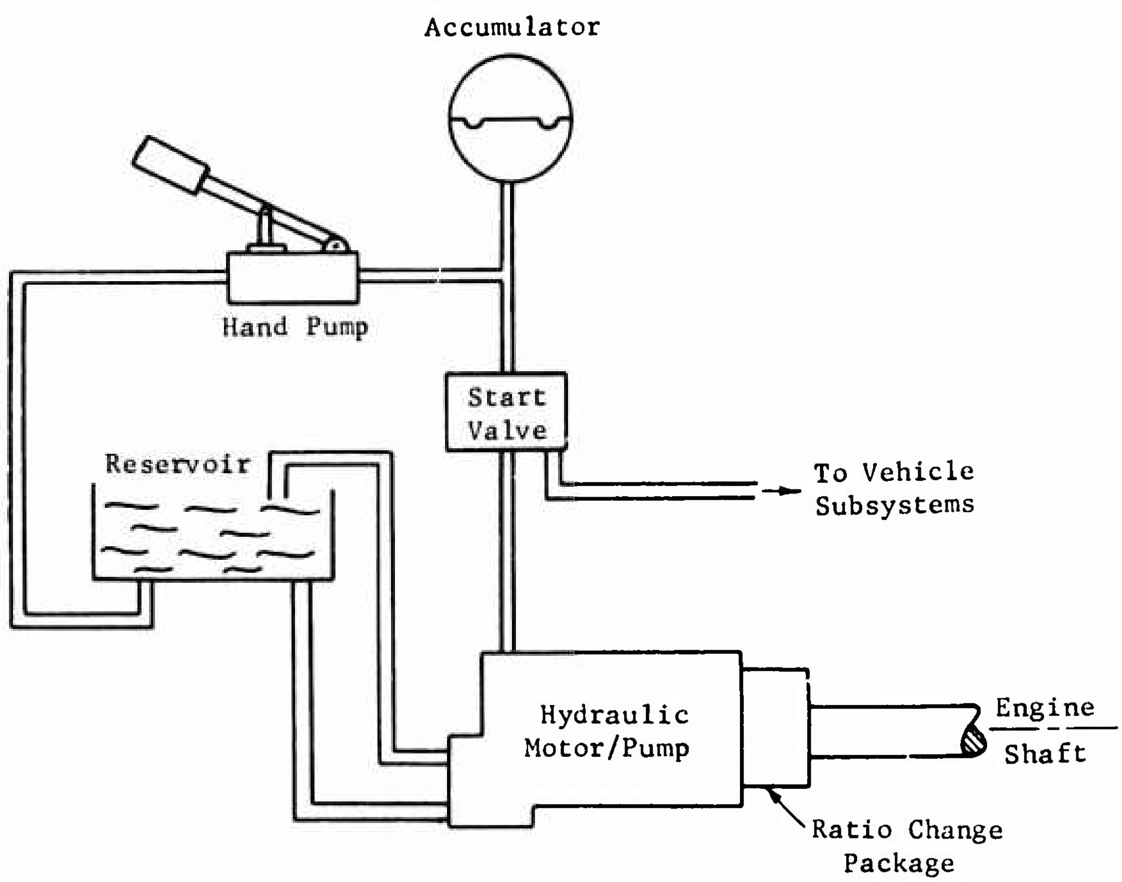


Figure 8. Thompson Ramo Wooldridge, Inc., Fuel-Draulic Starter System Schematic.



WEIGHT SUMMARY

Starter	2.0 lb
Accumulator	7.5 lb
Fluid	2.0 lb
Hand Pump	3.0 lb
Start Valve	1.0 lb
Lines & Misc. Hdw.	3.0 lb
Ratio Change Package	3.4 lb
Total	21.9 lb

Figure 9. Hydraulic Starter System Schematic.

During a start, this fluid is directed to a hydraulic motor/pump by way of the start valve. The details of the starter valve are shown in Figure 10. The torque developed by the hydraulic motor is applied through the locked up ratio change package directly to the engine rotor shaft. When the engine has started, the torque developed by the engine is transmitted to the motor/pump (now acting as a pump) through the same ratio change package. The ratio change package consists of a planetary gear system with two sprag clutches. This device is shown schematically in Figure 11. It is arranged so that a 4.8:1 speed reduction is obtained when the engine is driving the pump. This arrangement permits the motor/pump to operate up to 21,000 rpm during the starting cycle and at a maximum of 12,500 rpm during normal engine operation. These speeds are considered to be well within the capability of piston type motor/pumps of the size required.

As indicated, a hand pump is provided for use in the event of an aborted start. Approximately 100 pump strokes would be required to recharge the accumulator.

The characteristics of the starting system described here are shown in Figure 12. As indicated, the time required to crank the engine to cutoff speed is approximately 14 seconds. The system pressure at cutoff is 2000 psi.

The weight summary for the hydraulic starting system is shown in Figure 9. The system weight of 21.9 pounds is considered to be quite realistic if not conservative. The motor/pump weight is based on a typical fixed-angle pump which has a basic weight of 1.7 pounds. An additional 0.3 pound was allowed for the conversion of this unit to a motor/pump. The performance used is based on increasing the stroke angle of this unit from 30 degrees to 35 degrees. The basic pump unit supplied by the vendor is shown in Figure 13.

The accumulator weight is based on the assumption that the weight of an existing 7.5-inch-diameter accumulator could be reduced from 8.5 pounds to 7.5 pounds by the use of advanced fiber glass techniques. The weight of hydraulic fluid chargeable to the starting system is based on the use of 65 cubic inches of fluid per start. The reservoir and the remaining fluid are not charged to the starting system, since these are used to provide secondary power. A three-start system would weigh approximately 36.4 pounds. The increased weight includes an allowance for additional fluid and the replacement of the 7.5-inch-diameter accumulator with one 12 inches in diameter.

The hand pump weight was derived from catalog data. The remaining weights were estimated as part of the design study.

It is felt that the single-start system described above represents, in terms of current and near-future technology, an attractive approach to meeting the starting requirements of small gas turbine engines. The system, as outlined, not only provides for engine starting, but also provides secondary power in the amount of approximately 11.0 horsepower

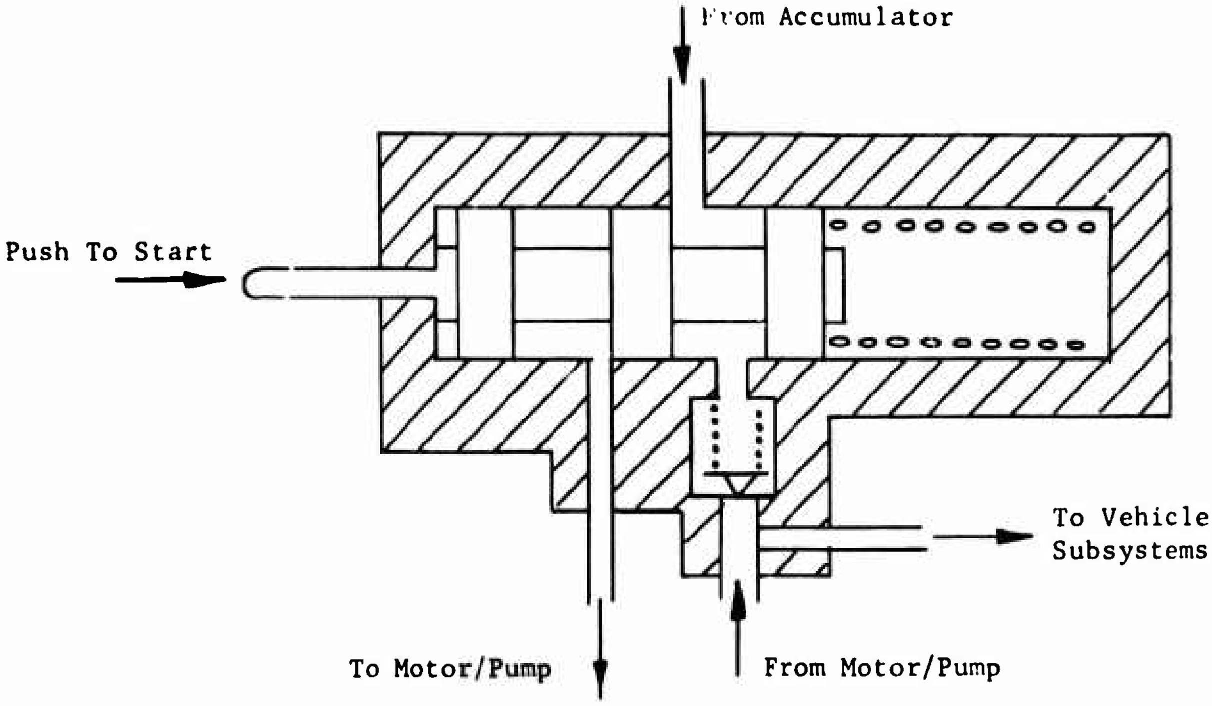


Figure 10. Hydraulic Starter Valve Details.

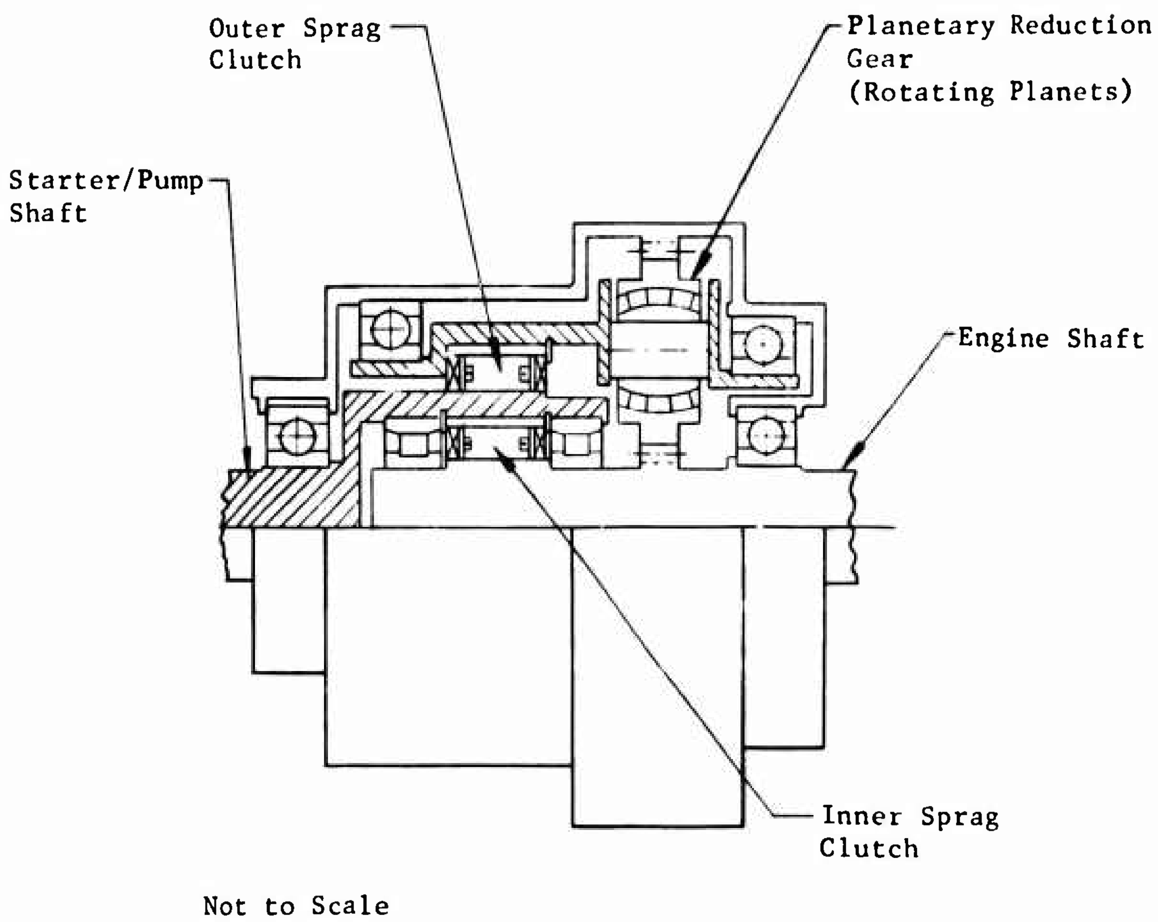


Figure 11. Hydraulic Starter Ratio Change Package.

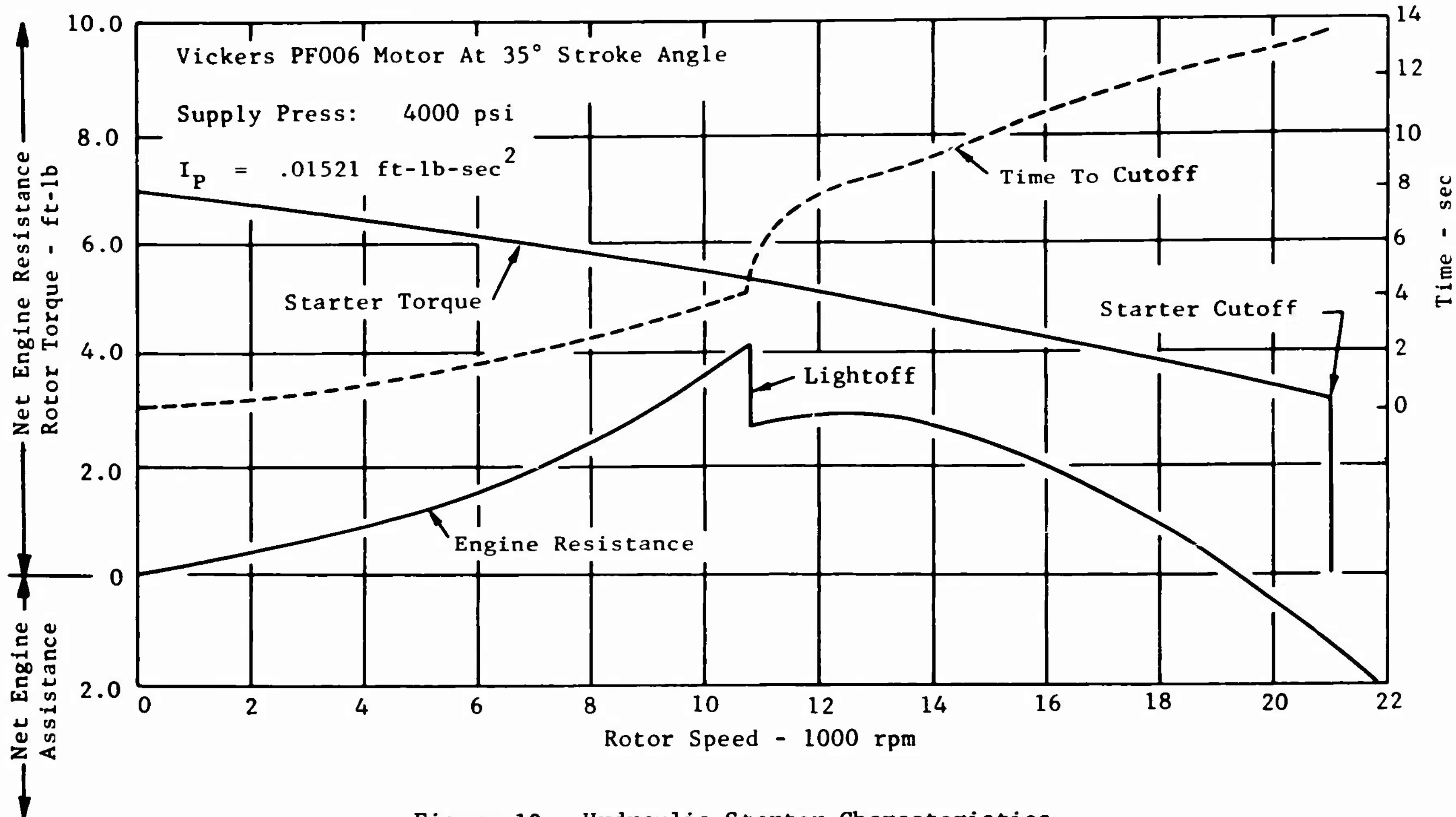


Figure 12. Hydraulic Starter Characteristics.

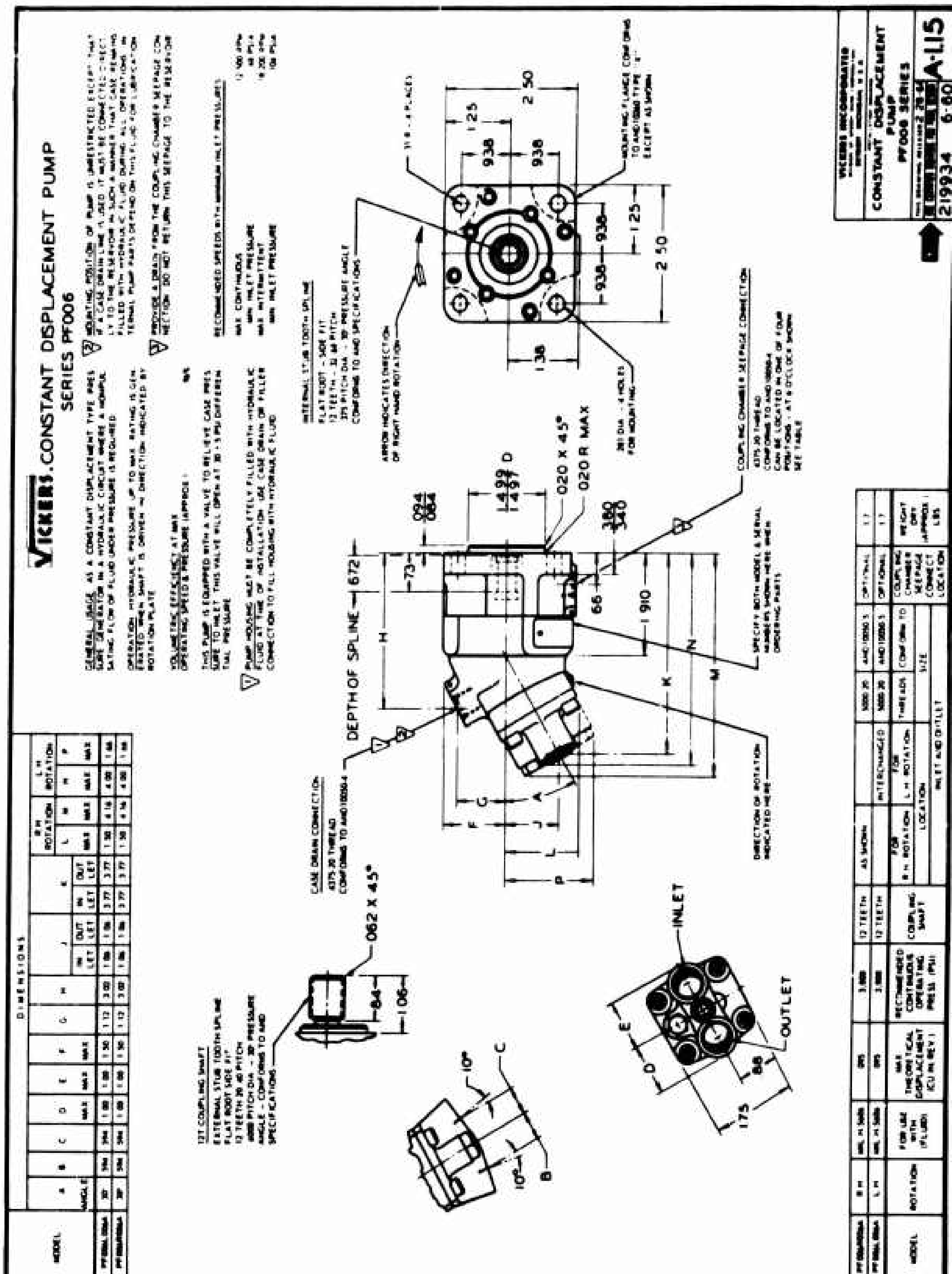


Figure 13. Vickers Hydraulic Motor.

at rated engine speed. The system is essentially self-sufficient in that repeated start attempts can be made with a minimum expenditure of manpower and time. Also, the system is composed of simple devices and, as such, should have a high order of reliability.

It is expected that a significant improvement can be made in terms of weight and size by continued development effort. However, it is doubtful that there will be drastic changes in the basic nature of the system. The two parameters that influence size and weight are subject to certain restraints. Rotating component speeds are not likely to reach the level desired, based upon comments received during this investigation. System pressure is likewise limited by practical aspects such as component stresses and hazard level.

It is recommended that the hydraulic starting concept be explored further as the most likely means of meeting the starting requirements for small gas turbine engines.

SPEED SENSING DEVICES

The Aeronautical Division of Honeywell submitted information on two types of fluidic speed sensors. The first, referred to as the analog speed sensor, provides an output differential pneumatic pressure that is proportional to rotational speed. This device makes use of a flat pickup head that is positioned adjacent to the periphery of a rotating cylinder. The hydrodynamic pumping action in the boundary layer between the pickup member and the rotating surface generates a positive pressure ahead of the tangent point and a negative pressure behind the tangent point. Pressure taps placed in the pickup head at the appropriate location give a differential pressure that is proportional to the rotational speed. This concept is illustrated schematically by Figure 14. The theory of operation of this device is explained in detail in Appendix III of this report. Honeywell is continuing the development of this device under an Air Force contract and feels that it has considerable potential for use in fluidic control loops. This contractor feels that serious attention must be given to the following in any specific application:

1. The suitability of the device relative to the environment in which it must operate; i.e., the adverse effects on performance caused by variations in temperature or contaminants of various types.

2. The complexity and reliability of a readout device that would be required for non-control use.

The second speed sensor concept presented by Honeywell is of the pulse type. In this device, pressure pulses are generated by means of a perforated or slotted disc which permits the pressure from a nozzle to be received at the pickup when a flow path is presented by the hole or slot. The pulse frequency is proportional to the speed of the rotating disc. This pulsating signal is used in conjunction with various fluidic elements in the computer and sequencing loops of the engine control. The same or similar signal could also be used to obtain a visual readout of speed.

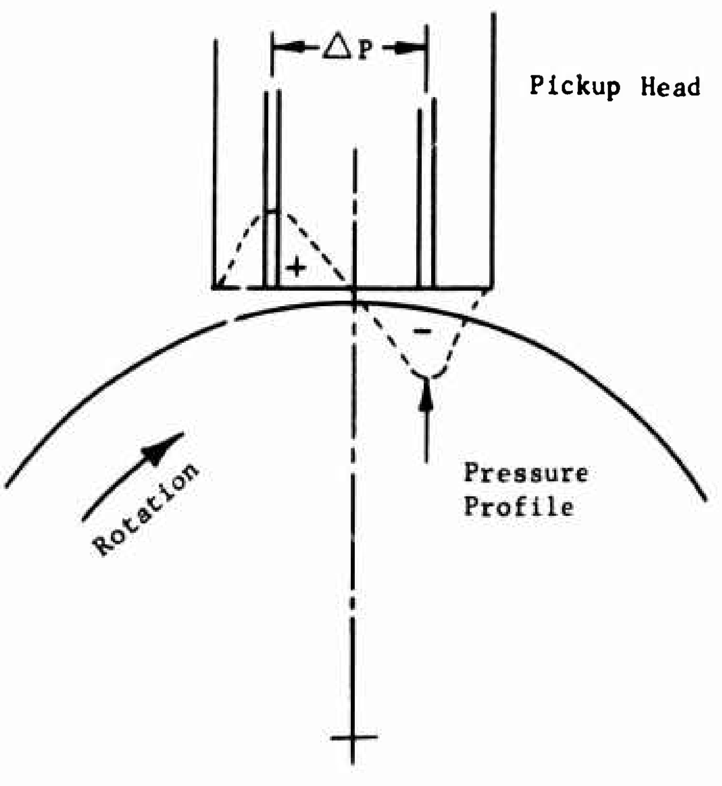


Figure 14. Analog Speed Sensor Schematic.

Honeywell claims, in the detailed discussion of the pulse type speed sensor in Appendix III, that a corrected speed signal, $N/\sqrt{\theta}$, can be obtained from this type of device. It is not obvious from their presentation that this is possible. Since θ is defined as the compressor inlet total temperature, T_{2_T}, divided by 520°R, it would seem that a corrected speed signal could be obtained only if compressor inlet pressure were used as the supply pressure for the speed sensor and associated bistable amplifier. Since compressor inlet pressure is not at an elevated level, it cannot be used as the supply pressure. In response to questions relative to the above, Honeywell has indicated that the compressor inlet temperature influence would be imposed on the delay line associated with the bistable amplifier. While such an approach is possible, it is the opinion of this contractor that the time response of the device would be poor. This would cause an error to exist between actual corrected speed and indicated speed during transient operation.

The pulse type speed sensor does not lend itself particularly to high speed applications. Honeywell indicates that the device is limited to a frequency response of about 200 cycles per second. The speed range associated with the small engines being considered would require a device capable of sensing frequencies in excess of 1600 cycles per second (100,000 rpm). Honeywell has indicated that additional fluidic elements can be employed to increase the frequency response of this type of speed sensor. It is felt that this additional complexity would not be warranted for a speed sensor used for direct readout. It may be required in order to meet the requirement of a particular control loop.

A rather detailed discussion is presented by Honeywell in Appendix III relative to various types of readout devices. From this discussion it would appear that if a relatively high order of accuracy is required and if the display is remote from the speed sensor, a hybrid fluidic/electronic approach is required. Before a specific readout concept could be selected, it would be necessary to investigate the requirements of a specific application and to conduct studies of the reliability, cost and availability aspects of the various elements of the systems. It would appear that in some instances it would be desirable to retain as much as possible of an existing readout instrument and adapt the fluidic or fluidic/electronic circuit characteristics to this device.

At this stage of development the role of the fluidic speed sensor used as a portion of a speed display circuit appears to be questionable. As will be indicated later in this section, there appears to be more direct means of obtaining a speed indication. It is expected that the development emphasis relative to fluidic speed sensors will be directed toward matching the characteristics of this type of device to the requirements of the engine control proper. For instance, one fuel control manufacturer* has indicated that the application of its version of an analog fluidic speed sensor to one of its existing pneumatic fuel

* Bendix Aerospace Division

controls would permit a significant reduction in size and weight and would eliminate the necessity to run the fuel control at less than engine rotor speed. This approach also opens up the possibility of a hybrid fluidic/hydromechanical fuel control in which the speed sensor and computer elements are separated for convenience of installation.

A second accessory manufacturer* points out, in information submitted relative to self-contained ignition systems and discussed in detail in the next section of this report, that a speed signal can be obtained in terms of voltage or frequency from a high speed ignition exciter. This signal can be used in conjunction with relatively simple circuitry and instrumentation to obtain a speed display.

Similar speed signals can also be obtained from the high speed special purpose generators that are manufactured by at least two other companies.** These devices were discussed on page 6 under Alternators and Generators. Here, again, the speed signal can be obtained at no cost in weight or space from a device that is serving some other additional purpose.

It is suggested that a speed signal for display purposes can best be obtained as a by-product from an electrical device and can be used with conventional readout devices. In some instances it may be possible to take advantage of a fluidic speed signal if the signal is already being generated for control purposes. In such a case it would be necessary to establish, by additional study, that the development of special readout equipment, if required, is warranted.

IGNITION SYSTEMS

The ignition system requirements for the 2-to-5-pound-per-second gas generator were limited to the specification of the output requirements only, in order not to limit the new concept types. These brief requirements are listed below:

Ignition Energy	2 joules
Spark Rate	2 sparks per second

To complete the overall ignition system specification, the manufacturers were requested to include a definition of the energy source, the igniter plug, electrical leads and any other equipment that may be unique and/or favorable to their system concept. The most desirable ignition system, from the standpoint of overall program objectives, would be one that is completely self-sufficient and does not require the conventional energy storage device or auxiliary electronics. Such a system would offer the additional advantages of providing a small amount of secondary power and of providing an engine speed signal.

* The Scintilla Division of Bendix

** R.E. Phelon Company and Varo, Incorporated

Two ignition system manufacturers provided information and described design concepts which might be applicable to small gas turbine engines.* The first manufacturer presented data to indicate their current capability. In the opinion of this contractor, the ignition system proposal by the second manufacturer is the most favorable from both conception and development possibilities. This system presents a favorable weight picture and has had considerable development background. Several systems have been designed and manufactured. In each case the design of the alternator was matched to the available engine mounting condition and the capacitor discharge ignition system desired for the application. A typical advanced ignition system including the alternator, the connecting lead to the exciter unit, the exciter unit, the two discharge leads, and the two spark igniters weighs a total of 2.35 pounds. It is also possible to provide a speed indication signal with this system of sufficient magnitude to eliminate the need for an engine-driven tachometer. The detail specification data for this system are provided below.

Number of Plugs Fired	2
Operating Altitude	10,000 feet
Ambient Operating Temperature Range	-65° +250°F
Spark Rate	4 sparks per second minimum at 2700 rpm
Stored Energy	1.55 joules, nominal
Duty Cycle	7 min on, 10 min off for 500 cycles
Output Voltage	3.5 kv, nominal
Max Alternator Speed	30,000 rpm

For all practical purposes, this system as specified can be considered directly applicable to the small gas generators under consideration. The manufacturer of this type of ignition system** has suggested and indicated several development approaches which are listed below for possible consideration.

Under certain conditions of engine operation, environment, and mission, significant further refinements can be made in ignition systems featuring an engine driven alternator, such as:

1. The possibility of developing an exciter unit/alternator combination that would create a high frequency ionizing pulse across the spark igniter electrodes permitting follow-through energy directly from the alternator. This would remove large components such as tank condensers and high energy spark gaps and permit a much simpler, longer lived type of ignition unit.

2. By suitable coordinated design, it appears feasible to make a glow plug that could be powered by the specially designed alternator without the need for any other black boxes in the system.

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* General Laboratory Associates and the Scintilla Division of Bendix

** Scintilla Division of Bendix

In either of the two approaches suggested, it is conceivable that the 2.35 system weight quoted above could be reduced to perhaps 1.5 pounds or less.

This contractor has utilized the above advanced concept in the engine installation study. without selecting a specific configuration. Such an approach permits a reduction in weight, size and complexity. If development effort is considered in this field it is the recommendation of this contractor that the overall engine light-off system, Figure 15, be considered for a self-sufficient system. Maximum use should be made of auxiliary power from the ignition alternator for battery charging, etc.

Engine scaling over a broader spectrum of engine sizes considered here should have a negligible effect, if any, on ignition system size, providing a high degree of commonality and reduction in production costs.

TEMPERATURE INDICATING SYSTEMS

The information received relative to temperature sensors and temperature indicating systems was submitted by Honeywell. Their temperature sensor is claimed to have a time constant one-tenth that of a compensated thermocouple. Such response characteristics are highly desirable. Only the unclassified details of the device have been presented, and Honeywell has indicated that additional information is available from the classified documents produced under Air Force Contract AF 33(615)-2696.

The fluid temperature sensor is a temperature-sensitive fluid oscillator. The hot gas, whose temperature is to be measured, impinges on a fixed splitter and is forced to oscillate between two cavities which provide a low impedance path for the hot gas. The cavity size sets the length of the path the gas will follow. Since the gas pulse will travel at the acoustic velocity, c, the oscillating frequency is a function of $\sqrt{T}$.

$$c = \sqrt{gKRT}$$

where g = acceleration due to gravity
K = adiabatic exponent
R = gas constant
T = absolute temperature of the inlet gas

Since K and R are known and are constant in the range of gas turbine engine operation, the frequency becomes a function of $\sqrt{T}$.

To be useful, the pneumatic frequency signal of the temperature sensor must be converted to a usable signal. Honeywell carries out this conversion through the use of a coupler which isolates the hot gas from the frequency discrimination system. The acoustic output of the sensor is fed into a resonator (frequency discriminator) which changes the variable frequency input signal into an acoustic output signal whose amplitude is a function of frequency. The resonator may be a tuned cavity of length equal to one-half the wavelength of the acoustic signal.

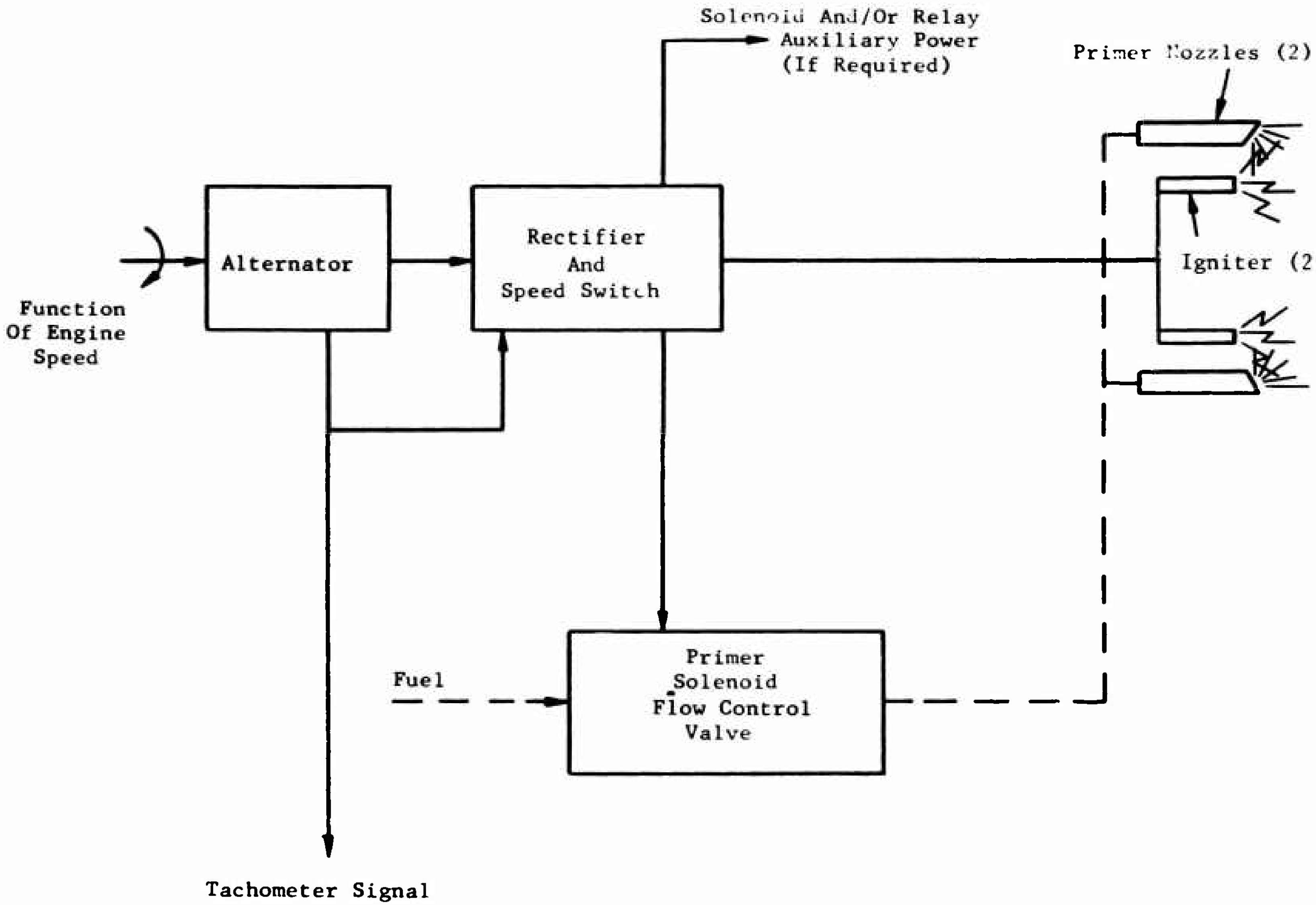


Figure 15. Primer/Igniter System Schematic.

Honeywell claims that in addition to the fast response of the fluidic temperature sensor, it has an ability to function in the elevated temperature environment at the turbine inlet. They suggest that the sensor be constructed of the same material as that used in the turbine. This approach is not considered to be completely satisfactory, since this contractor has found it necessary to use cooled turbine blades at the temperatures associated with advanced engines. It appears that much additional work must be done in order to establish a material that is suitable for use in that portion of the fluidic temperature sensing system that is exposed to the hot gases. It is anticipated that current high temperature materials development work being sponsored by USAAVLABS and others will be of considerable assistance in this area.

The output of the fluidic temperature sensing system can be used in the engine control loop or readout by one of the methods discussed in Appendix III of this report.

As a result of this program, it has been concluded that the fluidic temperature sensor concept has considerable merit and may provide a fast response device suitable for measuring temperatures in the range of 2500°-3000°F. However, it is felt that more emphasis should be placed upon the materials problem in order to insure the attainment of the above. Also, additional effort is required in order to establish a suitable arrangement for sampling the hot gas at the turbine inlet location with a minimum of air blockage.

During the course of the Wright Aeronautical study, USAAVLABS initiated a program directed towards the advancement of small, high temperature sensing devices (other than fluidic types) capable of measuring turbine inlet temperatures of 2500°F in a small gas turbine engine (2-to-5-pound-per-second airflow) which would be suitable for both temperature indication and incorporation into engine controls.

The results of this program should contribute greatly to the attainment of suitable controls for advanced gas turbine engines.

VARIABLE GEOMETRY ACTUATION DEVICES

There were no vendor inputs relative to actuation equipment. For this reason this contractor undertook a brief study to investigate the requirements for small engines. The first conclusion reached was that engine geometry would most likely be positioned either as a function of power lever position or in a manner related to corrected engine speed. To define this further would require a detailed knowledge of the specific engine and its application.

The design objectives of an appropriate actuation system must reflect the overall objectives of light weight, simplicity, and extreme reliability. It appears that these objectives would rule out complex electronic or

electrical servomechanisms. Even a conventional electromechanical actuation device would not satisfy the basic requirements due to the requirement for added electrical power and the need for relatively complex electrical circuitry.

It is suggested that the most promising approach lies in the adaptation of basic hydraulic servo-actuation practice.

A hydraulic servo-actuator of the type that might be used is shown schematically in Figure 16. Either hydraulic fluid or fuel could be used as the operating media. Fluid under pressure is supplied to the nozzle end of the actuator piston by way of an orifice. Fluid is also supplied to the differential area portion of the device at the rod end. A mechanical input displaces the cam follower and changes the flow rate through the nozzle. If the cam follower is displaced to the right, the flow rate through the nozzle cuts off momentarily and the pressure increases on the anti-rod end differential area causing the rod to extend. When the mechanical input has been satisfied position-wise, the flow through the nozzle becomes such that force balance is obtained on the two opposing differential areas.

A hydraulic servo-actuator of this type was used with success in a transmission control developed for Army Tank Automotive Center under contract DA30-069-ORD-3512. This actuator was found to have very good positional accuracy and stability. It is quite simple and therefore highly reliable. It is felt that a device of this type could be designed to meet most actuation requirements for small engines. While a mechanical input is shown, the device could be arranged to operate with electrical or pressure inputs.

CONSTANT SPEED DRIVE

In response to the Wright Aeronautical Division Problem Statement, one accessory manufacturer* submitted information covering a constant speed drive concept for small gas turbine engines. This manufacturer presented a discussion of constant frequency electrical systems and constant speed drives as related to aircraft requirements.

In view of the conclusions reached that a substantial amount of electrical power should not be extracted from the gas generator, there appears to be little need for a miniaturized constant speed drive.

Two points raised by the constant speed drive manufacturer are particularly significant to earlier discussions in this report relative to fluid pumps. The first is the conclusion reached by this company that centrifugal pumps of the size required are not feasible.

The second observation of this company relates to the speed limitations of positive displacement pumps. Their recommendations are that the speed of a 6-horsepower fluid pump should be limited to 15,750 rpm and that a

* Sundstrand Aviation

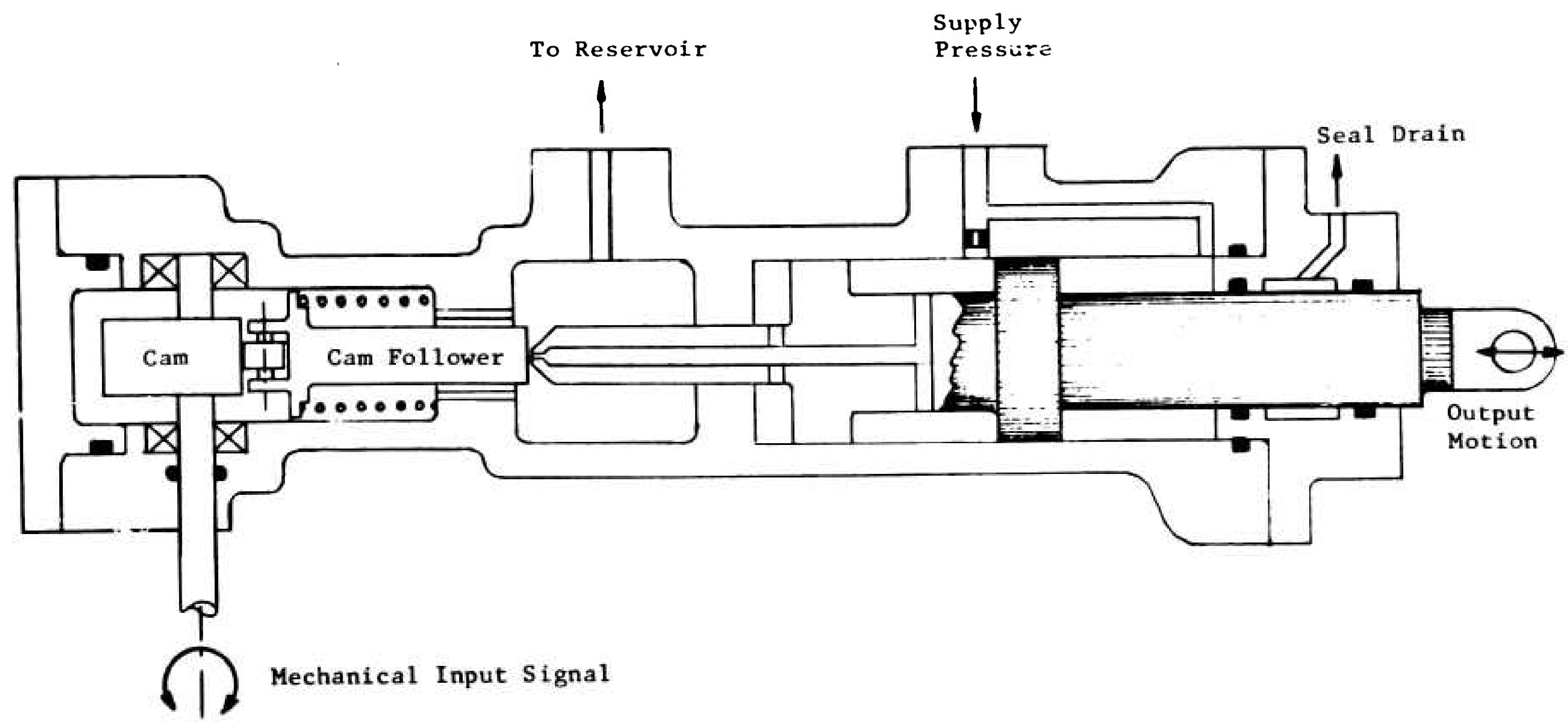


Figure 16. Hydraulic Servo-Actuator Schematic.

corresponding speed for a 16-horsepower fluid pump should be 11,400 rpm. The constant speed drive manufacturer's conclusions tend to confirm the recommendations of this contractor to the effect that advanced pump development should be directed toward design speeds in the range of 15,000 to 20,000 rpm.

ENGINE CONTROL SYSTEM

A significant input relative to engine control was obtained as a result of the Honeywell study effort. This work, conducted under subcontract to the Wright Aeronautical Division, was directed toward the application of fluidic technology to the control of small gas turbine engines. This complete program study is covered in detail in Appendix III of this report. Some comments, from an engine manufacturer's viewpoint, have been made in earlier sections of this report(Speed Sensing Devices and Temperature Indicating Systems), and discussion in this section will explore the overall control concept.

The control concept advanced by Honeywell is considered to be satisfactory with the use of a proportional-plus-integral governor providing good speed control. An acceleration fuel flow schedule based on turbine inlet temperature is, of course, a long-sought-after approach which, in turn, depends upon the availability of a highly reliable temperature sensing system. Comments relative to the potential of the fluidic temperature sensing system were presented in an earlier section of this report.

One problem, not specified for the Honeywell study, might result if the engine were surge limited in some areas of operation and temperature limited in others. This situation can be accommodated by the Honeywell approach through the use of a scheduled temperature limit.

There is no doubt that multifuel capability can be handled by a temperature limiting type of acceleration schedule. This is one of the major advantages of such a system. However, as noted before such an approach requires the availability of a highly reliable and responsive temperature sensing system.

The Honeywell engine control is presented in block diagram form in Figure 40 of Appendix III. A detailed discussion of the system is also presented in Appendix III and will not be repeated here. It should be noted that the absence of a fuel bypass loop implies the use of a centrifugal type fuel pump. Since centrifugal pumps do not scale well to the level required for small engines, a positive displacement pump will be used and a bypass loop will become necessary. This point was discussed with Honeywell and appropriate changes were made in the metering valve envelope to accommodate a bypassing type metering head regulator.

Additional refinements might be required in the design of the fuel metering valve to provide a positive fuel cutoff and to provide a windmilling bypass arrangement. These details would vary according to the application.

Some comments on the Honeywell engine/control simulation are in order. The results generally indicate that satisfactory engine operation can be obtained with the control mode and selected method of implementation. However, it would appear that more rapid accelerations could be obtained by optimizing the shape of the temperature limiting line as suggested by Honeywell. This is especially true in the 80-to-100-percent speed range

where the results indicate rather poor engine response. Reference to Figure 52 in Appendix III indicates that a very large increase in acceleration fuel flow is possible, particularly above 70-percent speed. Failure to take advantage of this resulted in engine response characteristics which are the reverse of those normally expected. Generally, engines respond faster in the higher speed regime. It is felt that adequate response can be obtained by the proper matching of acceleration fuel flow to the engine surge and temperature limits. In some types of engines it may be possible to exceed the normal temperature limit by a considerable margin during a transient condition to improve response further. In such a case, the transient would be of such short duration that the critical engine parts would not experience a significant change in temperature.

The cyclic action of the temperature limiter is not desirable, but it is difficult to state with certainty that unsatisfactory engine operation would result from the resulting low amplitude fluctuations in pressure and temperature. It is expected that a more satisfactory temperature limiter characteristic could be obtained by reasonable modifications to the control circuit should one find this necessary.

The Honeywell Start and Relight Sequence Control concept is discussed in detail in Appendix III. It is agreed that many desirable functions are accounted for by this system. However, the additional complexity represented by the total system proposed may not be warranted except for special applications. It is also suggested that the starting fuel flow scheduling portion of the overall system be provided as an element of the engine fuel control, since such a basic function is performed by this circuit.

Honeywell has assumed that supply pressure for the fluidic elements can be obtained from the air supply used for starting the engine. Since there is no assurance that an air starter will be used, it is evident that additional effort is required to establish a suitable pressure supply for the fluidic control when compressor discharge pressure is inadequate.

Comments on the Honeywell fluidic speed sensors and the fluidic temperature sensor were presented in earlier sections of this report.

The Cockpit Speed Control described by Honeywell performs the required function but may be sensitive to contaminated air, since it makes use of a slot that is .003 inch wide. Any change in the pneumatic calibration of such a device with time would make for unsatisfactory engine operation.

As noted earlier, the fuel metering valve discussed by Honeywell in Appendix III is not directly applicable in a bypassing type metering loop. This has been corrected subsequent to the preparation of the final report by coordination with Honeywell, which resulted in the addition of a bypass valve in the engine installation study.

The Honeywell fuel control system is shown schematically in Figure 17. The installation of the Honeywell components is discussed in a later section of this report.

The estimated weights as provided by Honeywell are as follows:

Speed Sensor (Analog and Pulse Type)		0.73 pound
Temperature Sensor		0.15
Fuel Control Computer		0.60
Fuel Metering Valve		1.55
Metering Head Regulator		0.60
Air Filter		0.58
Pressure Regulator		0.98
	Total	5.19 pounds

It is recognized that the weights quoted by Honeywell are estimates and that some reduction in weight might result from an intensive development program. It is interesting to note a weight comparison between the Honeywell system and a pneumatic control currently in use on small gas turbine engines.* The current weights are:

Main Fuel Control		4.0 pounds
Temperature Compensator		0.3
Air Filter		0.58
	Total	4.88 pounds

The existing control, shown schematically in Figure 18, makes use of fly-weights for a speed input. It has been indicated that by applying a particular version of a fluidic speed sensor, a significant reduction in weight and volume is possible. This modification will also eliminate the need to reduce the speed input to a fraction of engine shaft speed.

It can be concluded from the Honeywell effort that the fluidic control concept is basically applicable to small gas turbine engines. However, it is felt that much additional work is required in order to clarify and substantiate claims relative to:

1. Weight
2. Volume
3. Cost
4. Reliability
5. Contamination Sensitivity
6. Performance

* Courtesy of Bendix Aerospace Division

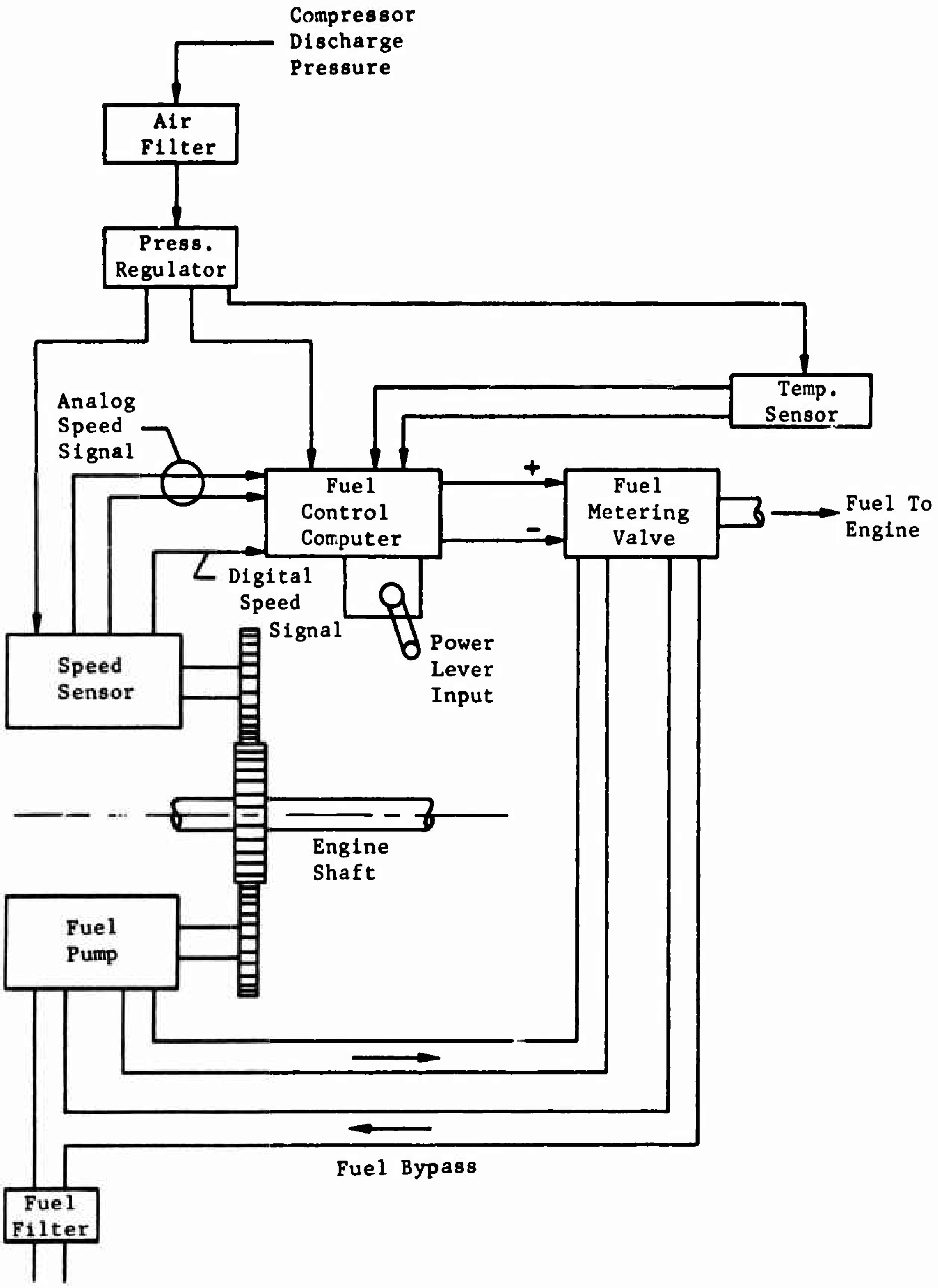


Figure 17. Honeywell Fuel Control Schematic.

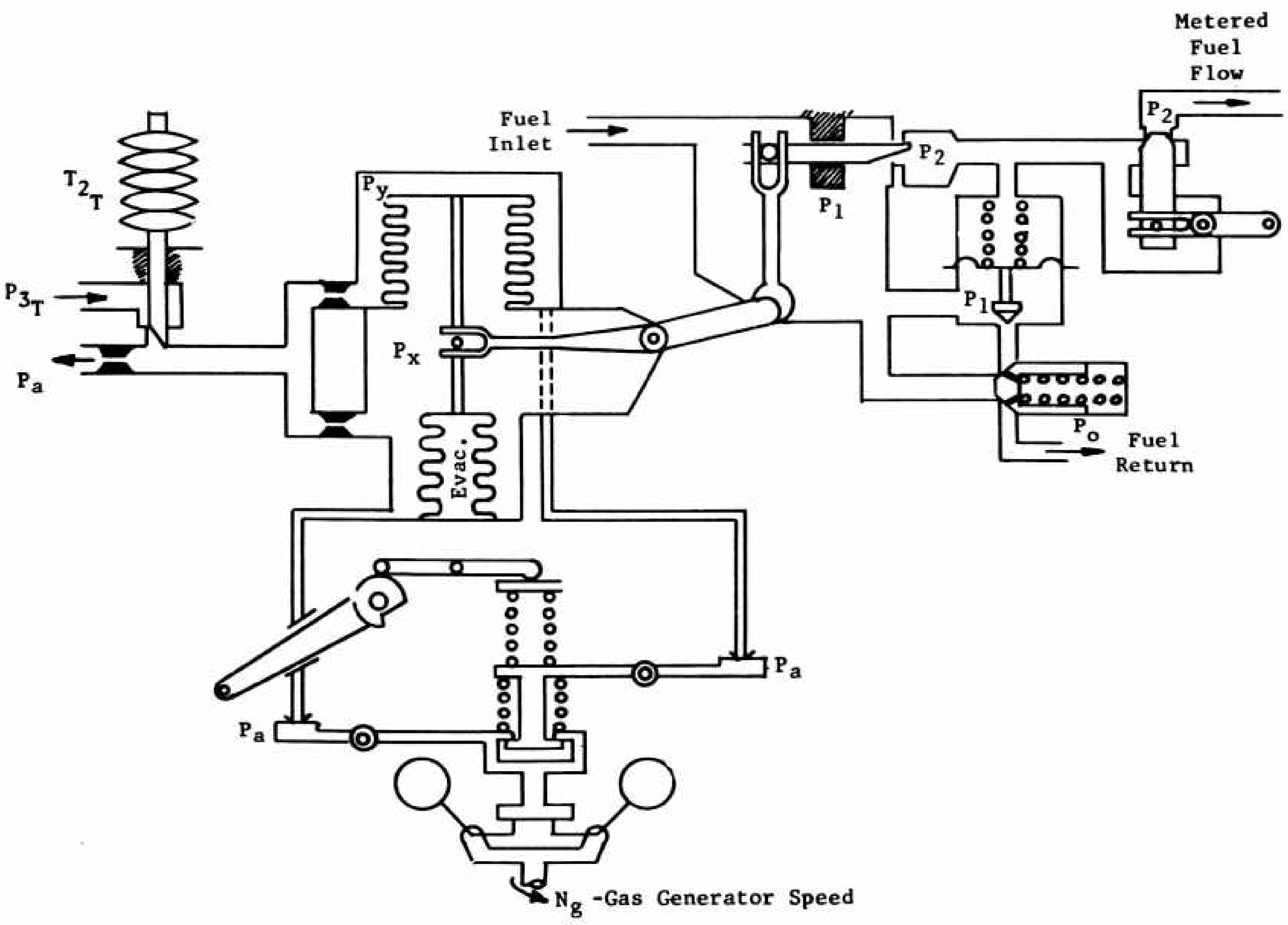


Figure 18. Bendix DP-A Series Fuel Control Schematic.

In addition, it appears that there is a need for more practical experience with fluidic elements of the type proposed before the true potential of these devices can be established.

The Honeywell analysis of the engine control requirements and the subsequent analog simulation are considered adequate under the conditions imposed by this program. However, the method of analysis and the engine time constant derived indicate the need for substantiation by experimental test. The analytical derivation of the engine time constant implies that the influence of the polar moment of inertia of the rotating elements of the engine does not scale directly from larger engines. A brief investigation of the physical characteristics of some small engines would indicate that this is true. However, it is suggested that this be explored experimentally in detail when small engines of the type considered are available. The resulting information will be of considerable value to the control designer.

The reliability aspects of a fluidic system remain to be established. It is felt that much additional work must be accomplished in the areas of contaminant resistance and environmental sensitivity. It is granted that fluidic elements should be insensitive to acceleration and vibration effects.

The weight aspects of the comparison of a fluidic system to a more conventional system are not convincing, as will be pointed out later. This is not surprising since the fuel metering elements, which defy scaling, will always constitute a significant percentage of the total system weight. It could be argued that a hybrid electronic/hydromechanical control system would be very competitive from the weight standpoint.

The Honeywell reference to a system based upon the use of a compressor surge sensor is interesting but questionable. All effort to date has indicated that oncoming surge is difficult, if not impossible, to detect. It has also been shown that once surge is encountered a drastic reduction in fuel flow is required in order to recover. The Honeywell effort in this area should be followed with interest.

The cost aspects of a fluidic control are also difficult to assess at this time. It is felt that many basic problems remain to be solved before cost can be considered seriously. Commonality of parts and production rates, of course, will have a significant bearing on this aspect of the problem. Honeywell states that fluid technology is in the development stage.

One aspect which is not emphasized by Honeywell, but which is considered significant, is the possibility of redundancy. It is doubtful that a hydromechanical or conventional fuel control can be miniaturized to the extent that would permit redundancy in the critical elements. Such might be the case with the fluidic approach. It is hoped that in some particularly critical applications, such as V/STOL aircraft, the fuel control could obtain the benefits of redundancy and self-monitoring circuits. However, here again, a hybrid electronic/hydromechanical system may offer the same possibilities.

INSTALLATION OF ENGINE CONTROLS AND ACCESSORIES

The engine used for the installation study is an advanced design that is being studied by Curtiss-Wright. This design features high turbine inlet temperature and high compressor pressure ratio. The design airflow is 4 pounds per second and the maximum speed is 60,000 rpm. The gas generator section of the engine is estimated to weigh 51.0 pounds.

Two basic approaches were followed in the controls and accessories installation. In both, emphasis was placed on achieving the most compact and serviceable arrangement for the components that must be shaft driven. In the first approach a conventional drop gearbox was designed to accommodate the various driven accessories. The gearbox is configured for easy removal from the engine, and the gearing is sized to provide the following component speeds at maximum engine speed:

Hydraulic Motor/Pump	direct driven up to cutoff (21,000 rpm)
Lube Pump	15,000 rpm
Scavenge Pump	15,000 rpm
Fuel Pump	15,000 rpm
Speed Sensor	35,000 rpm
Ignition Exciter	35,000 rpm

These component speeds were selected on the basis of the results of this program as reported earlier in this report.

The gearbox arrangement is shown in Figure 19. As shown, the gearbox is driven by a quill shaft from the engine rotor. Conventional gearing is used to achieve the various accessory drive ratios.

The gearbox as shown is estimated to weigh 11.65 pounds. A weight-reduction study conducted relative to this design indicates the possibility of a reduction in weight to 9.32 pounds.

The use of this gearbox in a complete engine installation is shown in Figure 20. Here, the rotating components are mounted as before, and the other controls and accessories are mounted on the engine structure. In this drawing the Honeywell fluidic control components are shown. The size and weight of the fluidic control elements were provided by Honeywell. The envelopes and weights for the fluid pumps were established by this contractor. The motor/pump and the ignition exciter dimensions were derived from vendor data. The non-driven components could be relocated, if desired, to accommodate a specific application. For instance, if these components were mounted on the upper half of the engine, better protection could be afforded against small-arms fire. The location of the fluidic temperature sensor is, of course, fixed by the function of this device.

The weight summary presented in Figure 20 indicates a total weight of engine mounted controls and accessories of 25.57 pounds. With the potential weight reduction this value would be approximately 22.5 pounds. This total includes the weight of the gearbox and hydraulic motor/pump ratio change package.

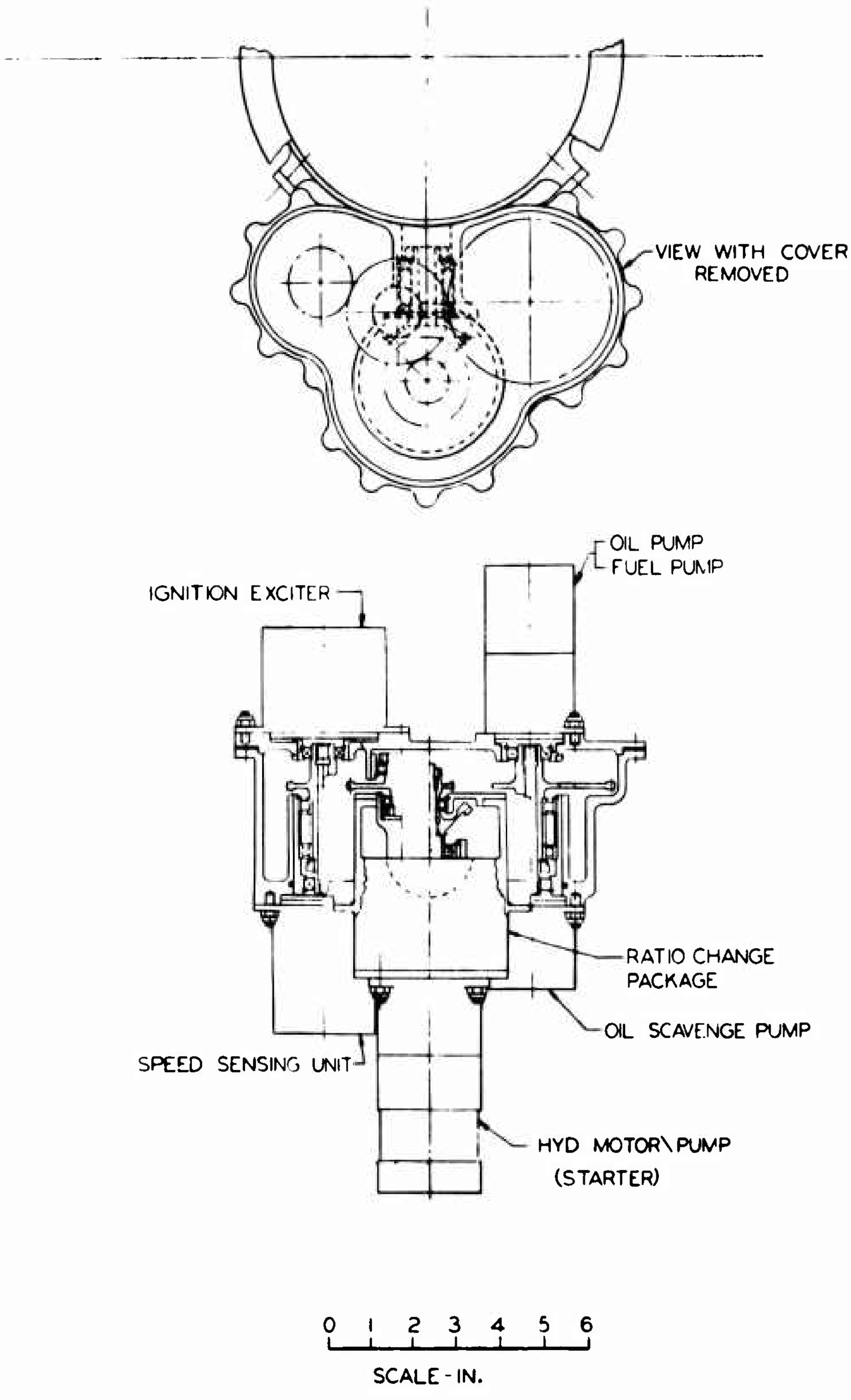


Figure 19. Accessory Gearbox, Drop Configuration.

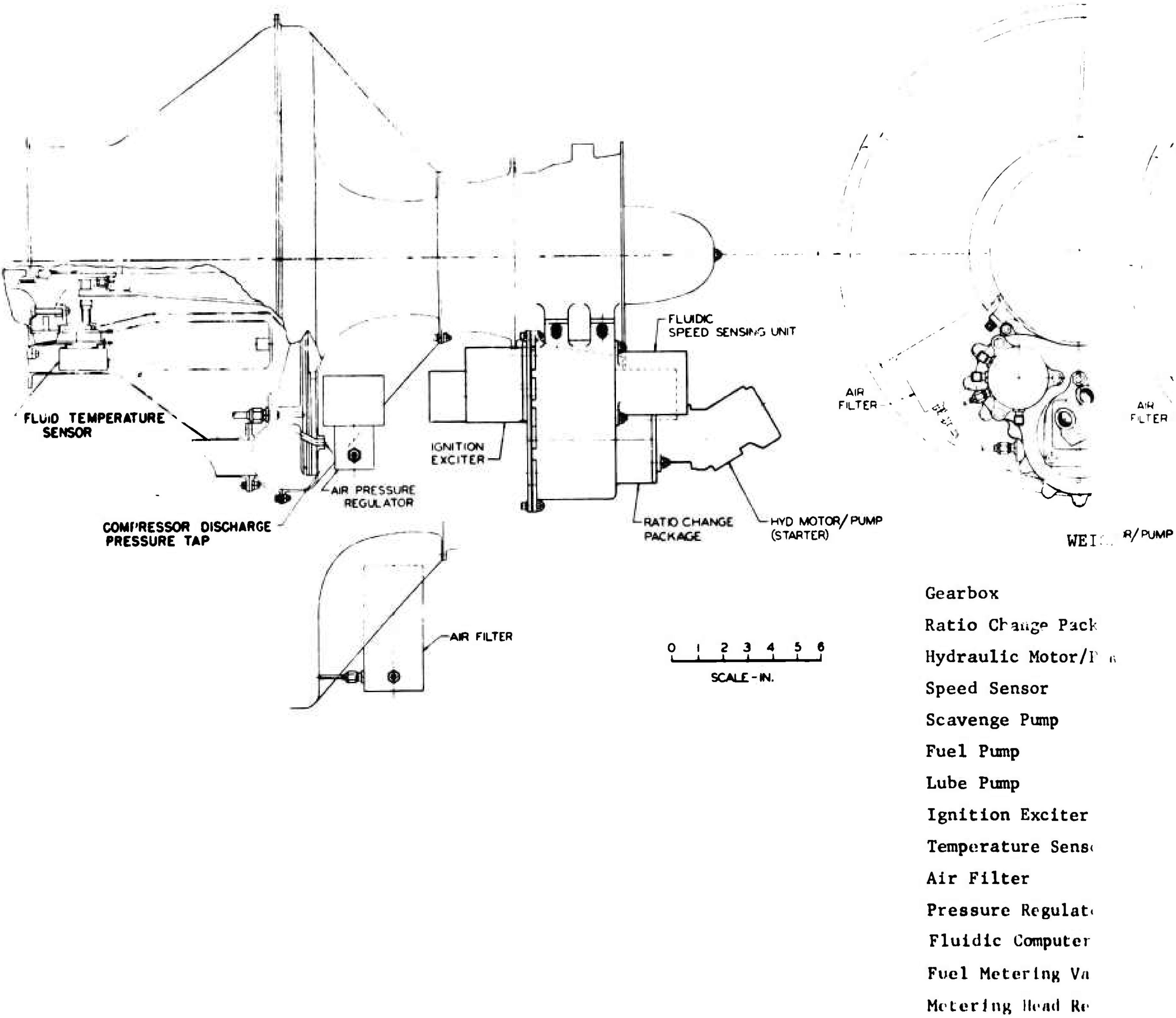
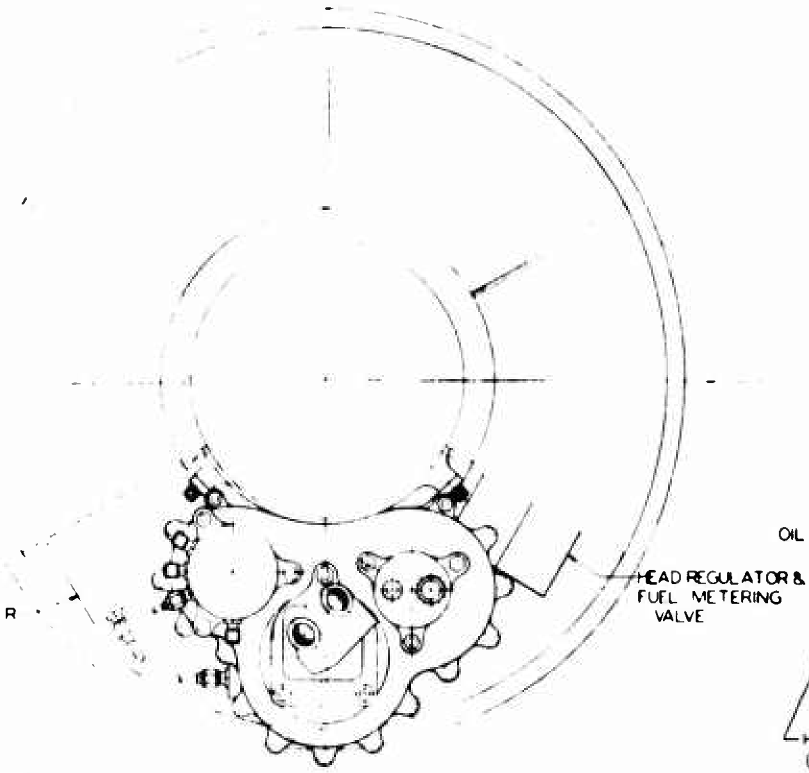
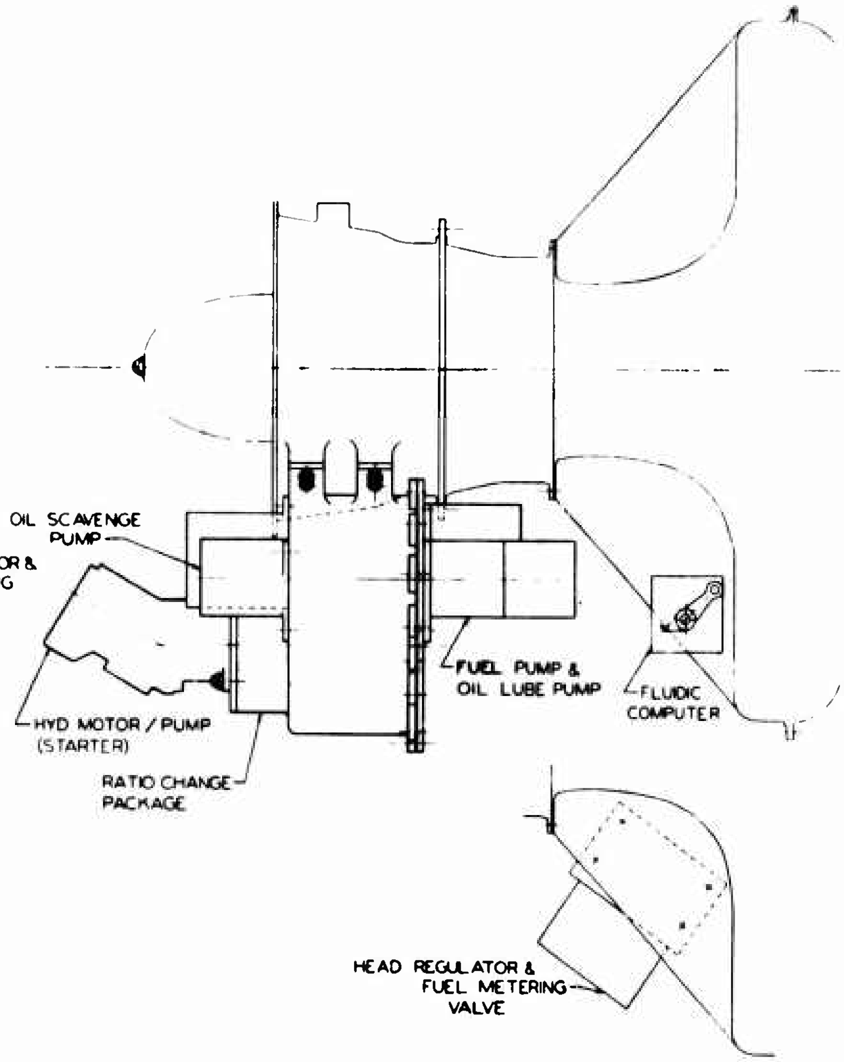


Figure 20. Engine Installation With Drop Gearbox, Honeywell Control.





WEIGHT SUMMARY

Gearbox	11.65 lb
Ratio Change Package	3.43 lb
Hydraulic Motor/Pump	2.00 lb
Speed Sensor	0.73 lb
Scavenge Pump	0.60 lb
Fuel Pump	0.60 lb
Lube Pump	0.60 lb
Ignition Exciter	1.50 lb
Temperature Sensor	0.15 lb
Air Filter	0.58 lb
Pressure Regulator	0.98 lb
Fluidic Computer	0.60 lb
Fuel Metering Valve	1.55 lb
Metering Head Regulator	0.60 lb
Total	25.57 lb

B

While the weight obtained by the use of the Honeywell fluidic components is attractive, it is felt that the total volume required can be improved upon. Such an improvement is shown in Figure 21 in which the Honeywell control elements are replaced with a conventional pneumatic control.* This type of control has been used on the T63 and PT6 engines. With the conventional control all functions are contained in the same housing, with the exception of inlet temperature compensation.

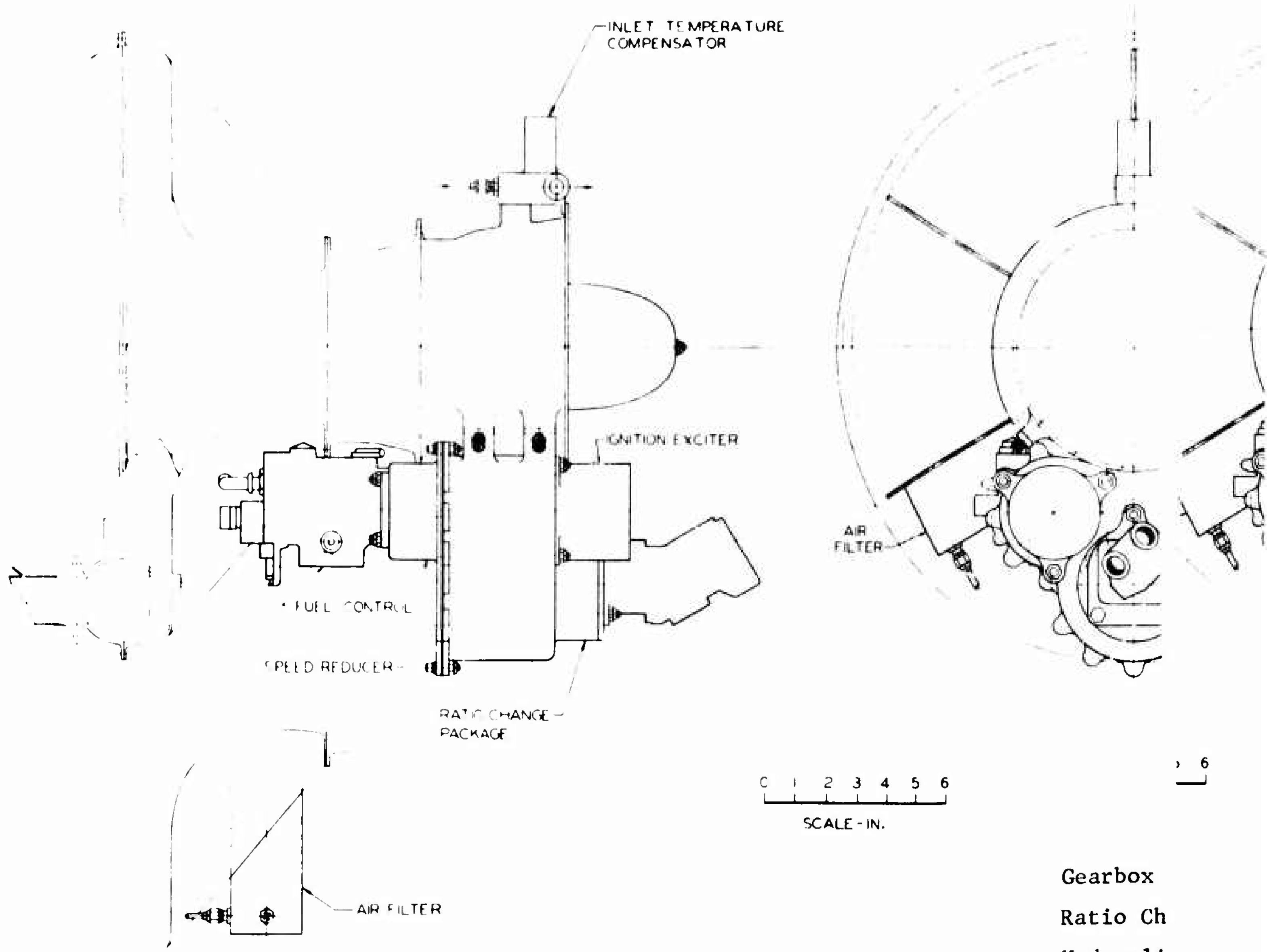
As shown in Figure 21, an allowance has been made on the gearbox for a speed reducer to accommodate the requirements of the conventional pneumatic fuel control. As mentioned earlier in this report, the control manufacturer has indicated that the requirement for a speed reduction can be eliminated by the adaptation of their version of a fluidic speed sensor to the pneumatic fuel control. This modification would also permit a size and weight reduction in the basic fuel control. If it is assumed that the modification suggested by the control manufacturer is made, the 2.5-pound allowance for the speed reducer would be eliminated and the fuel control weight would be reduced to 3.0 pounds. This would reduce the total engine accessories and gearbox weight from 27.76 pounds to 24.26 pounds, making this arrangement slightly more attractive than the one shown in Figure 20. If the anticipated gearbox and ratio control package weight reduction is applied, the potential weight of the arrangement using fluidic speed sensor is 21.4 pounds.

In order to verify the selection of hydraulic starting as the most promising approach, the installation drawing shown in Figure 22 was prepared. This drawing shows the installation of a suitable electrical starter of the type currently available. Not only do the size and weight of the starter represent direct disadvantages, but the size of the starter has undue influence on the size and configuration of the gearbox. Since the other driven components must be moved outward to accommodate the starter, the whole package becomes excessively large. The estimated weight of this configuration is 57.06 pounds. Also, as indicated in the section on electrical starters, there is a large weight penalty associated with the batteries required for this system.

The second accessory drive arrangement is shown in Figure 23. Here, an unconventional approach is taken with several objectives in mind. It is believed that a significant weight and space saving can be achieved if the rotating components are arranged so that their housings become part of the gearbox. This is achieved by making the pumps, etc., in cartridge form.

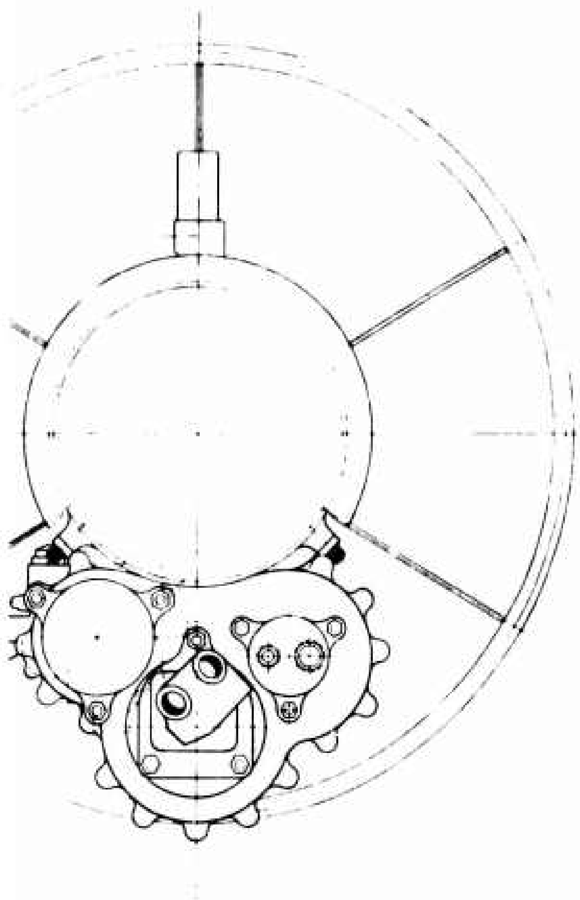
The configuration shown is designed to eliminate all external plumbing forward of the gearbox parting surface. All fuel, oil, and air passages are cored or drilled in the gearbox housing. These passages are manifolded and carried aft to service attachments on the engine as indicated in Figure 23. With such an arrangement the gearbox can be removed or installed by unskilled personnel with no possibility that lines can be mismatched or

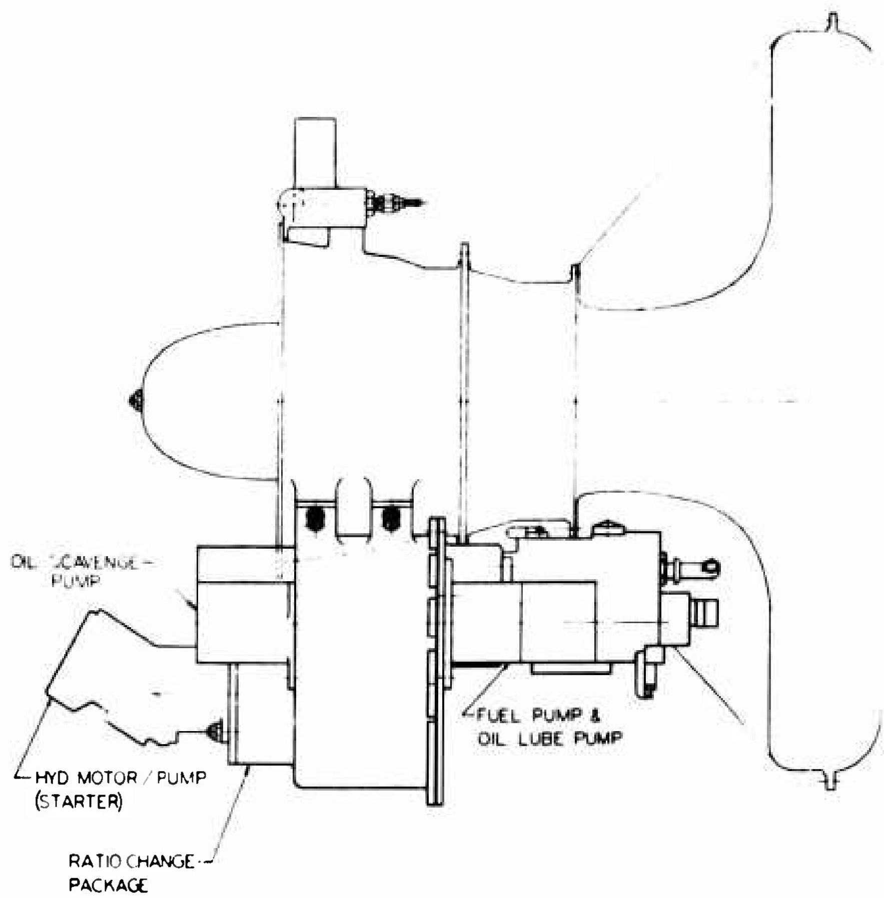
* Control envelope and weight courtesy of the Bendix Aerospace Division.



Gearbox
Ratio Ch
Hydrauli
Scavenge
Fuel Pum
Lube Pum
Ignition
Temperat
Air Filt
Fuel Cont
Fuel Cont

Figure 21. Engine Installation With Drop Gearbox, Bendix Control.

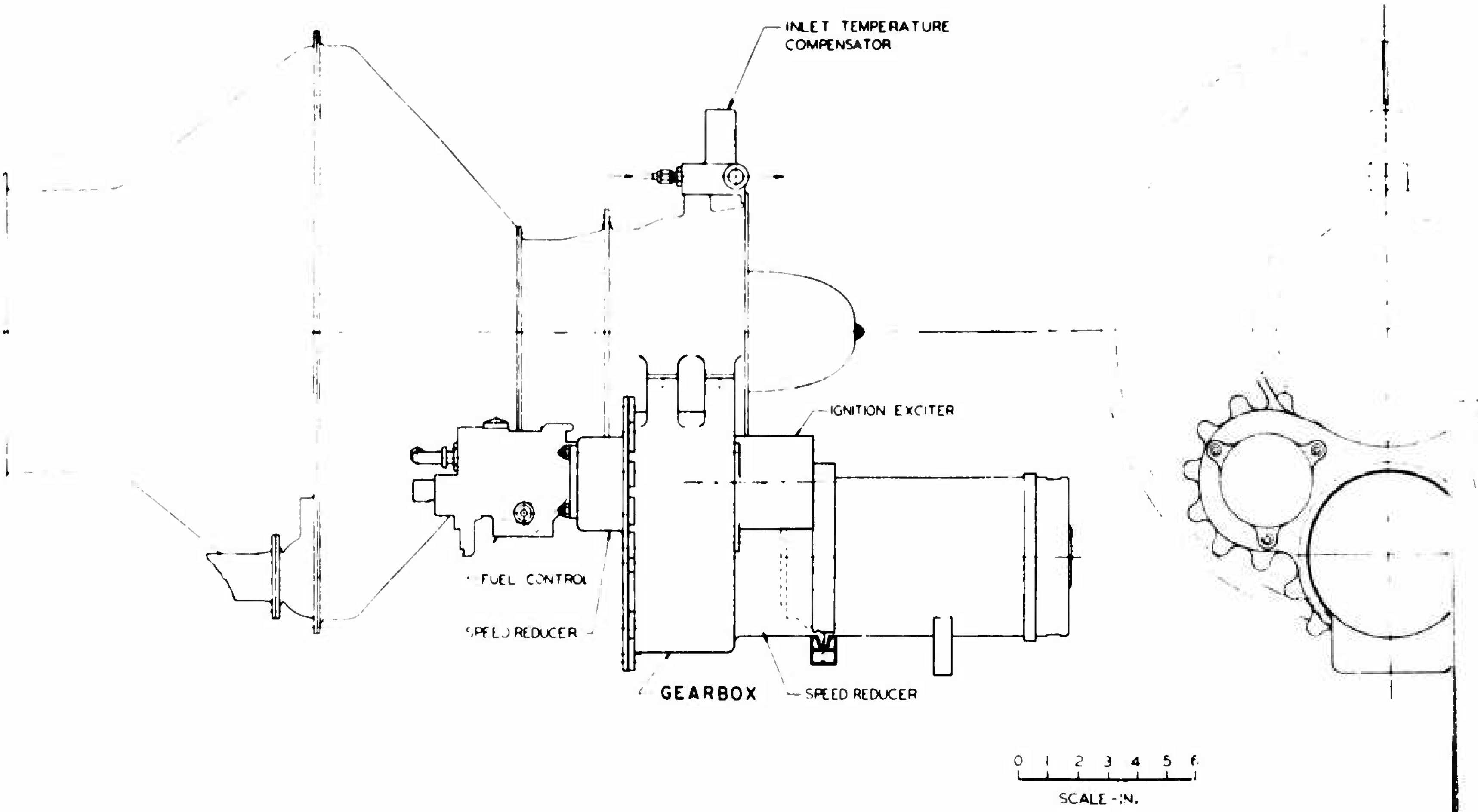




WEIGHT SUMMARY

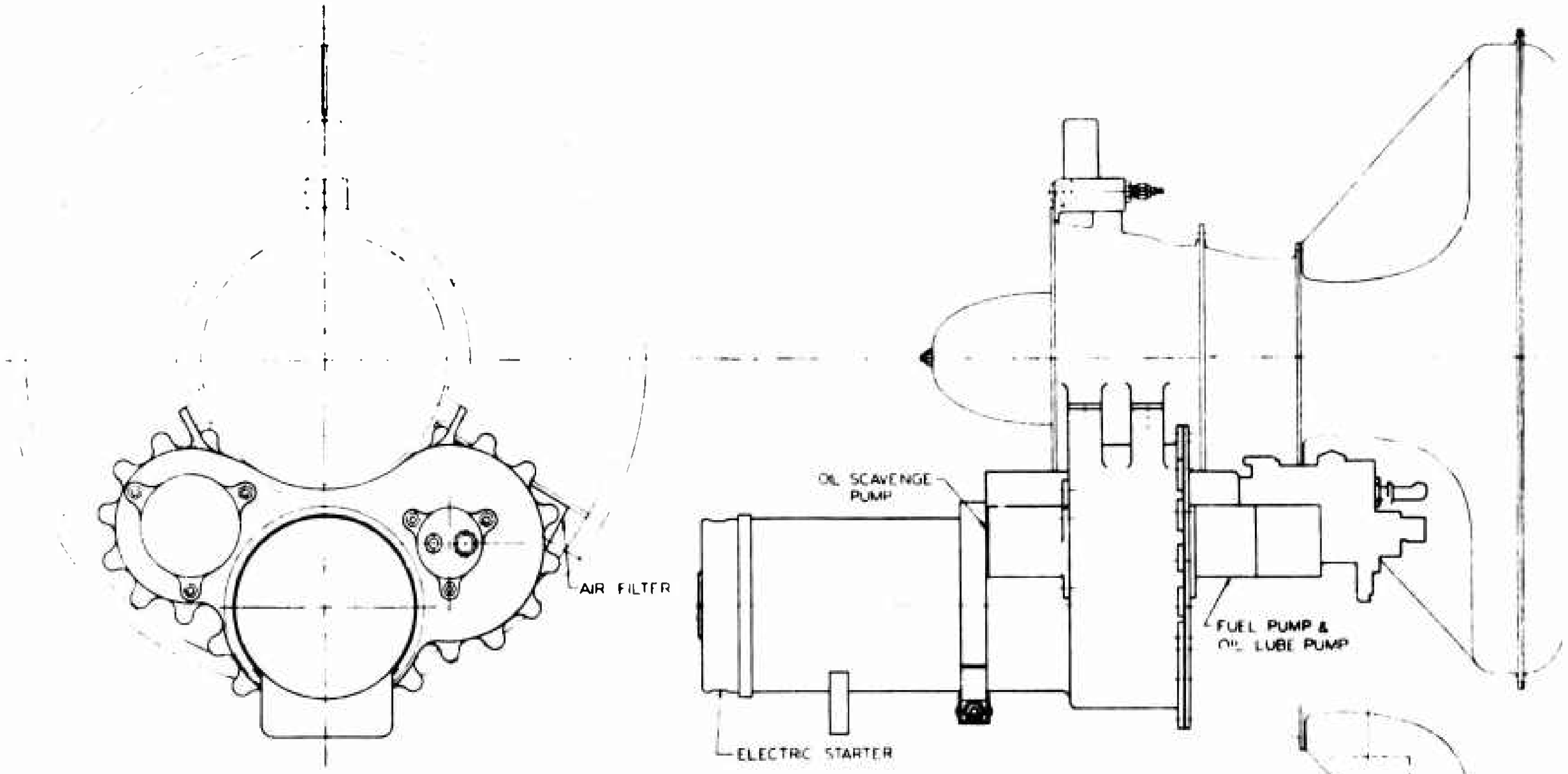
Gearbox	11.65 lb
Ratio Change Package	3.43 lb
Hydraulic Motor/Pump	2.00 lb
Scavenge Pump	0.60 lb
Fuel Pump	0.60 lb
Lube Pump	0.60 lb
Ignition Exciter	1.50 lb
Temperature Compensator	0.30 lb
Air Filter	0.58 lb
Fuel Control	4.00 lb
Fuel Control Speed Reducer	2.50 lb
Total	27.76 lb

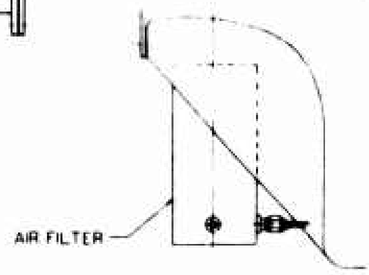
B



Gea
Ele
Spe
Sca
Fue
Lub
Ign
Tem
Air
Fue
Fue

Figure 22. Engine Installation With Drop Gearbox, Electrical Starter.





WEIGHT SUMMARY

Gearbox	15.88 lb
Electric Starter	26.50 lb
Speed Reducer for Starter	4.00 lb
Scavenge Pump	0.60 lb
Fuel Pump	0.60 lb
Lube Pump	0.60 lb
Ignition Exciter	1.50 lb
Temperature Compensator	0.30 lb
Air Filter	0.58 lb
Fuel Control	4.00 lb
Fuel Control Speed Reducer	2.50 lb
Total	57.06 lb

B

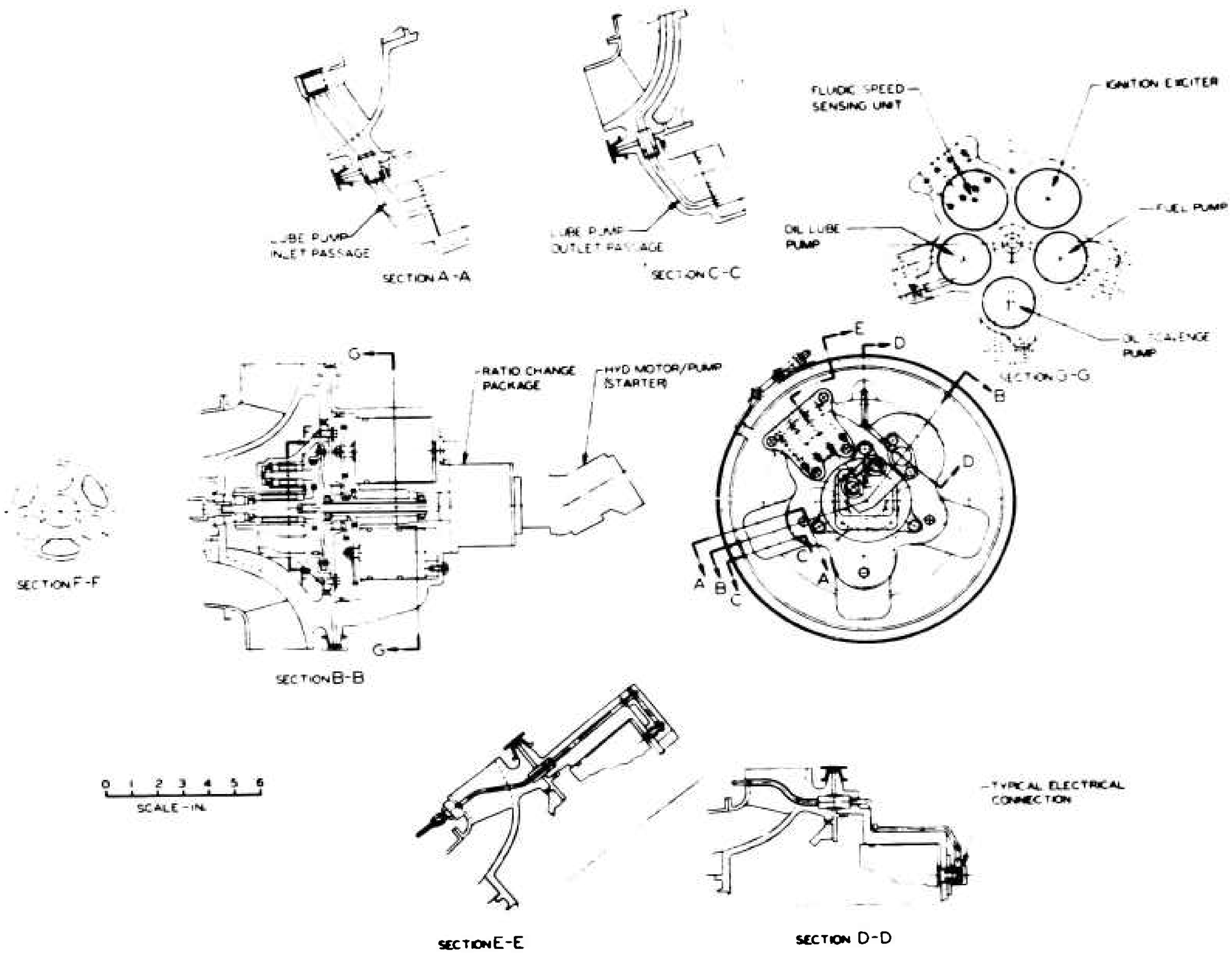
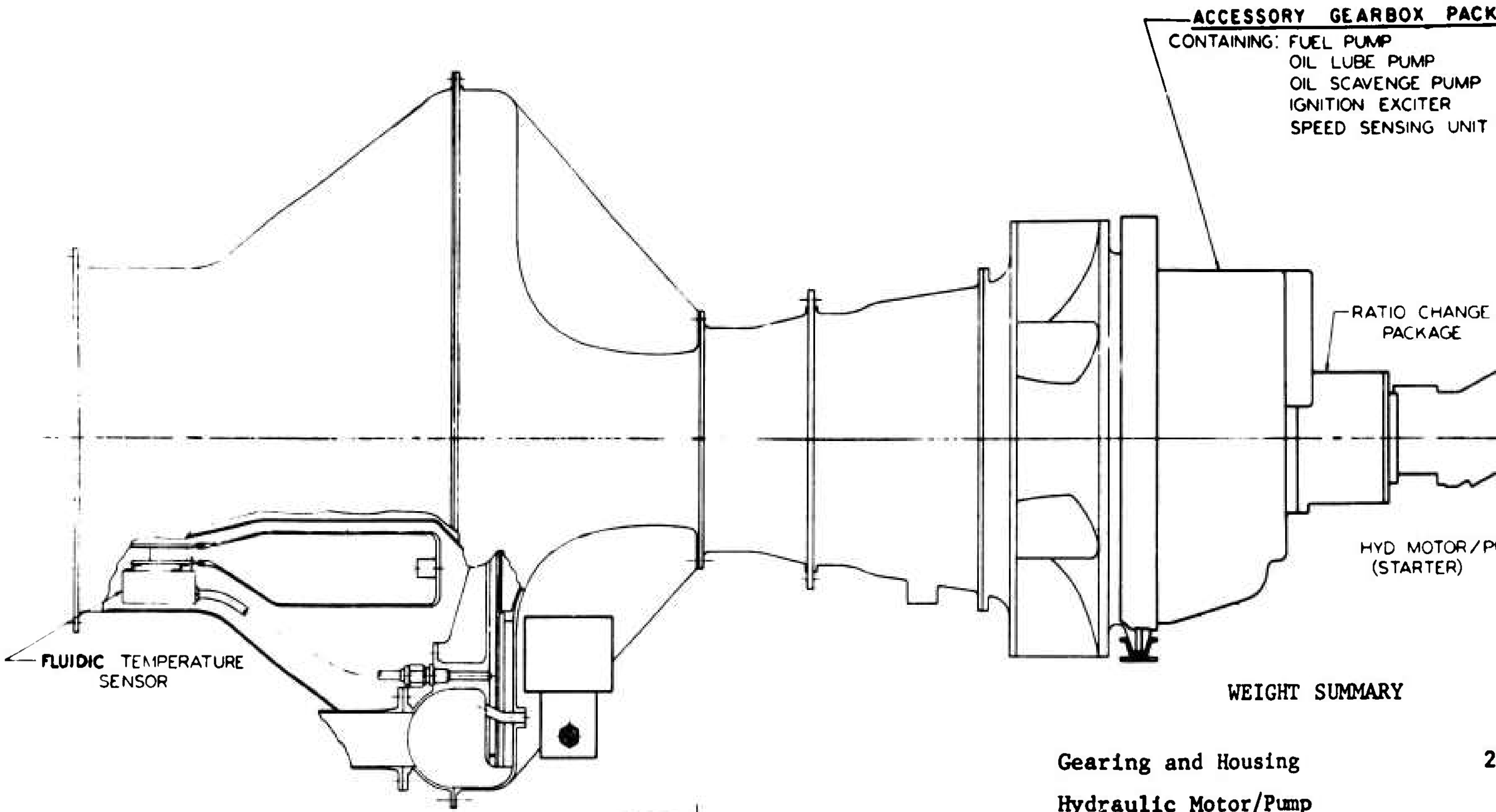
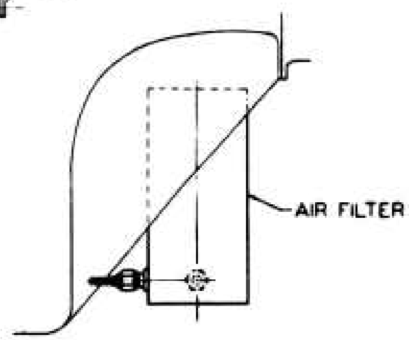


Figure 23. Accessory Gearbox, Nose Configuration.



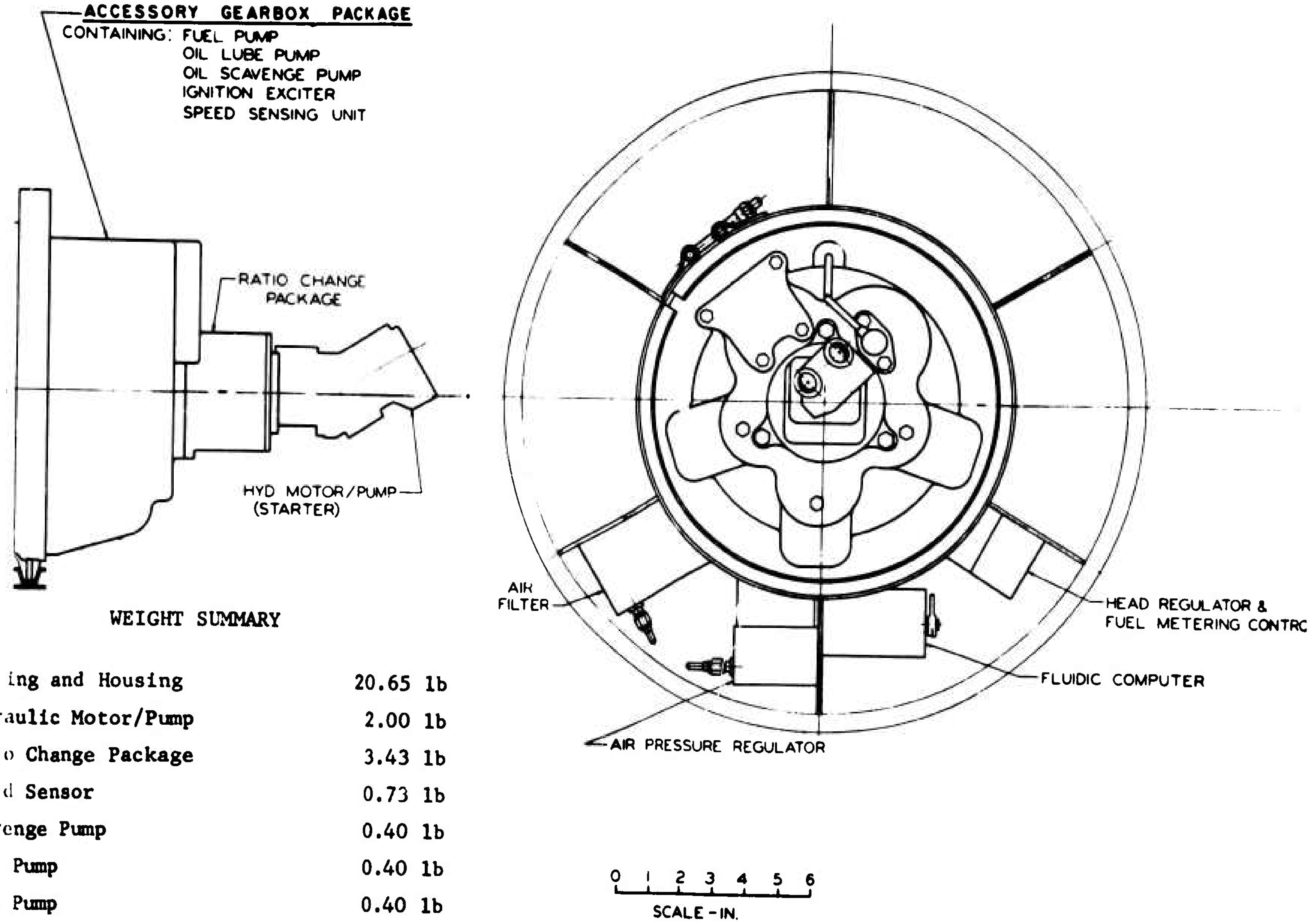


WEIGHT SUMMARY

Gearing and Housing	2
Hydraulic Motor/Pump	
Ratio Change Package	
Speed Sensor	
Scavenge Pump	
Fuel Pump	
Lube Pump	
Ignition Exciter	
Temperature Sensor	
Air Filter	
Pressure Regulator	
Fluidic Computer	
Fuel Metering Valve	
Metering Head Regulator	
Total	3

Figure 24. Engine Installation, Nose Gearbox.

A



WEIGHT SUMMARY

ing and Housing	20.65 lb
aulic Motor/Pump	2.00 lb
o Change Package	3.43 lb
d Sensor	0.73 lb
enge Pump	0.40 lb
Pump	0.40 lb
Pump	0.40 lb
tion Exciter	1.50 lb
rature Sensor	0.15 lb
ilter	0.58 lb
sure Regulator	0.98 lb
ic Computer	0.60 lb
Metering Valve	1.55 lb
ing Head Regulator	0.60 lb
Total	33.97 lb

B

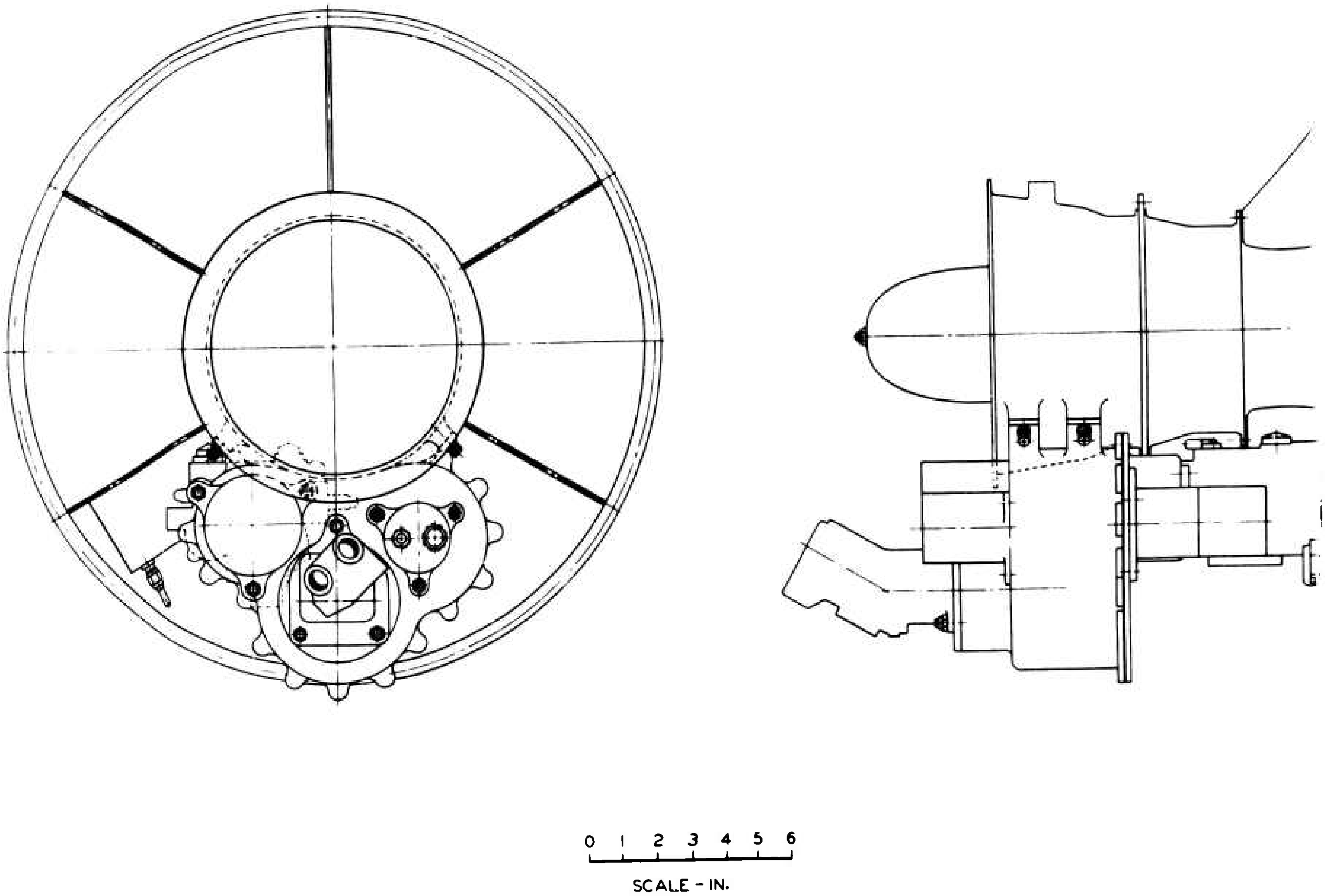
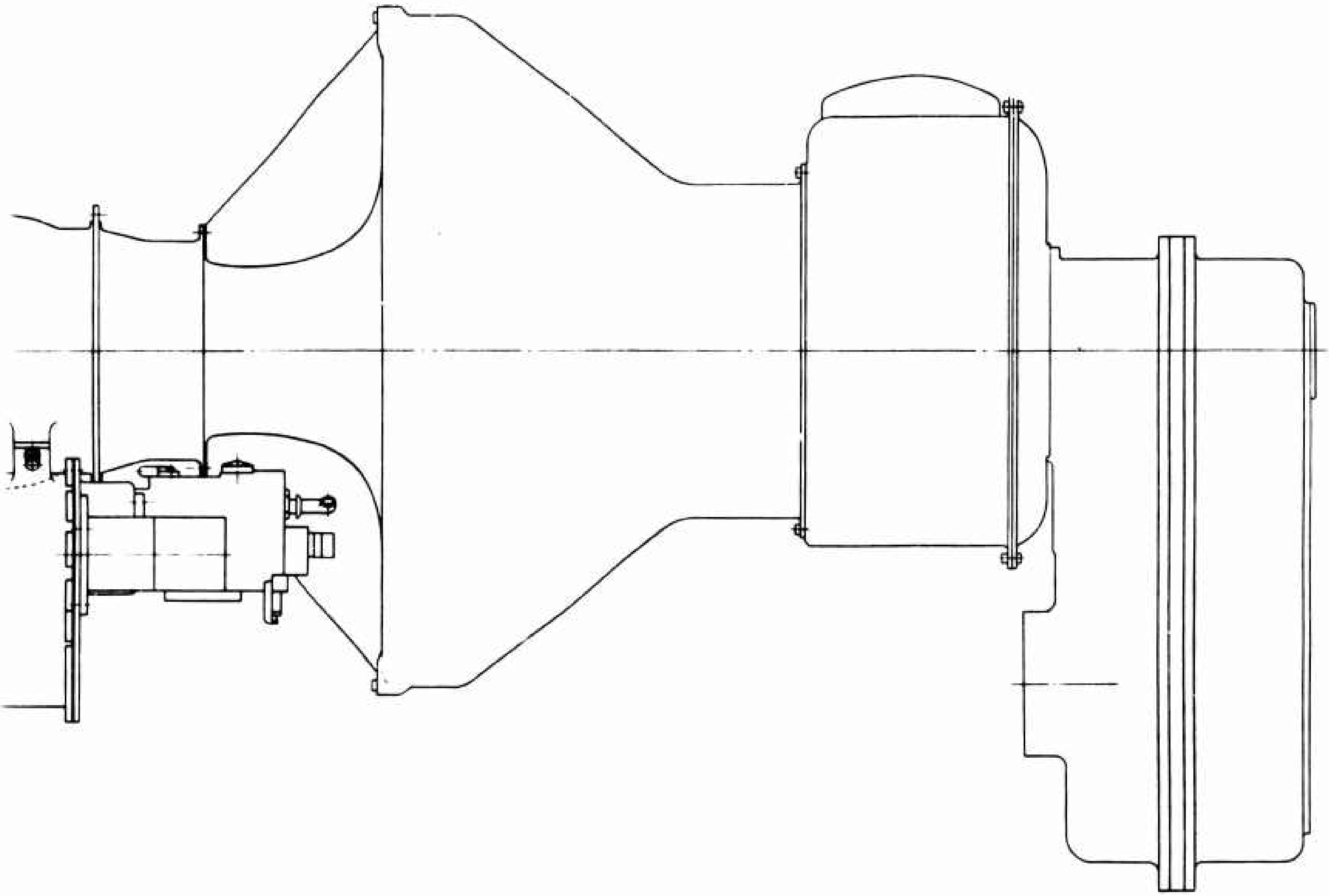


Figure 25. Turboshaft Engine With Accessory Gearbox.

A



B

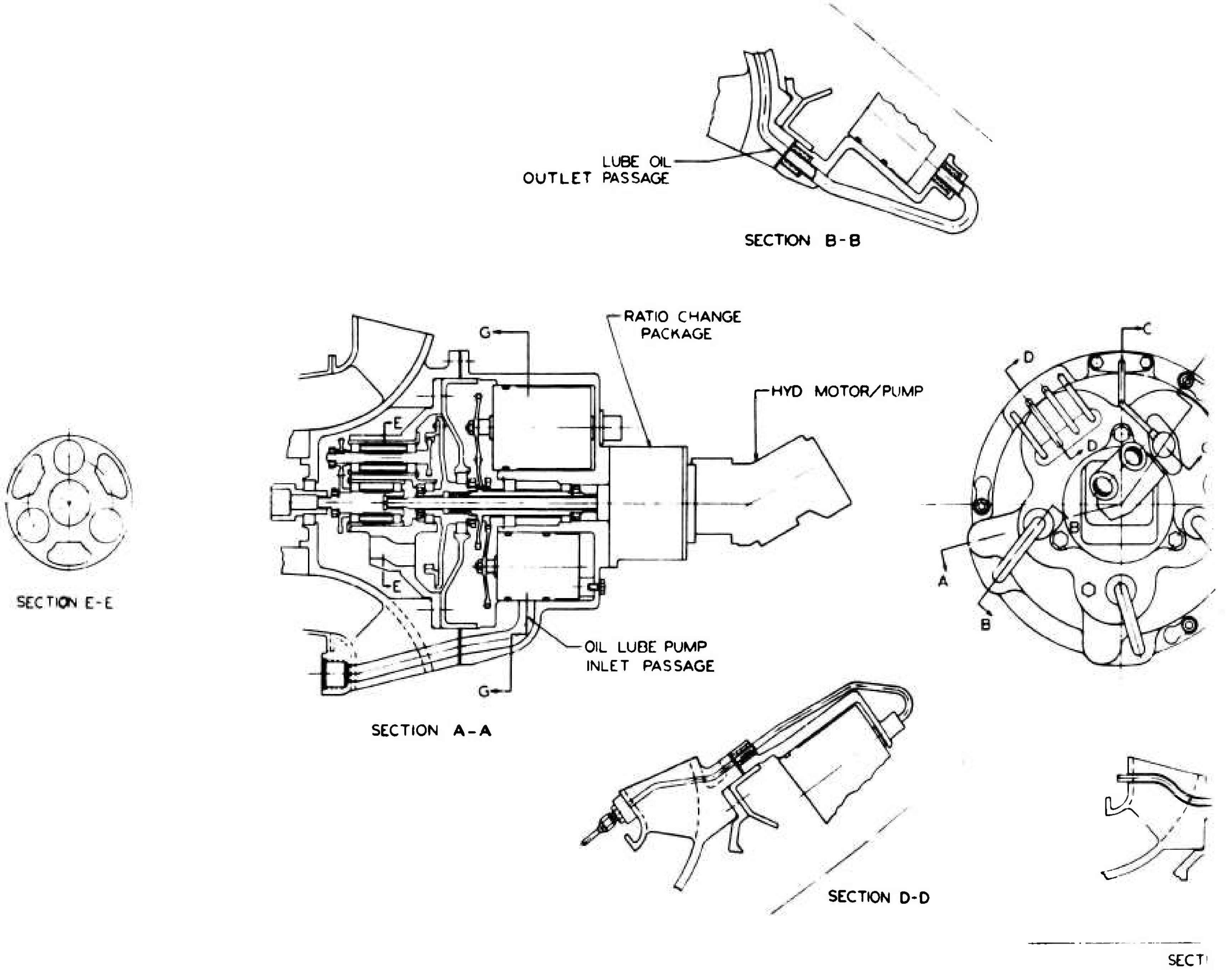
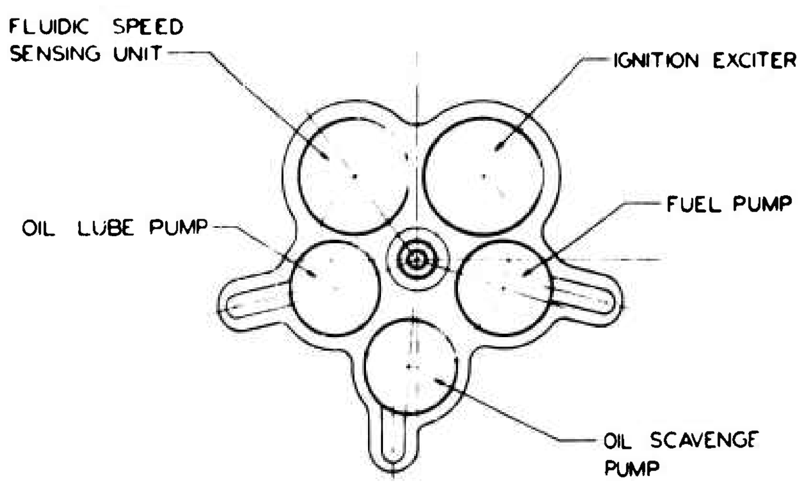
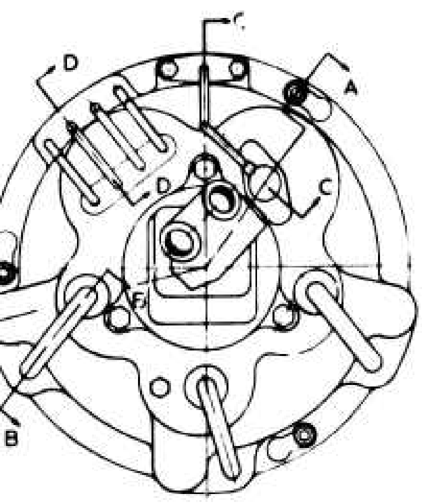
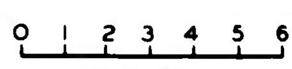


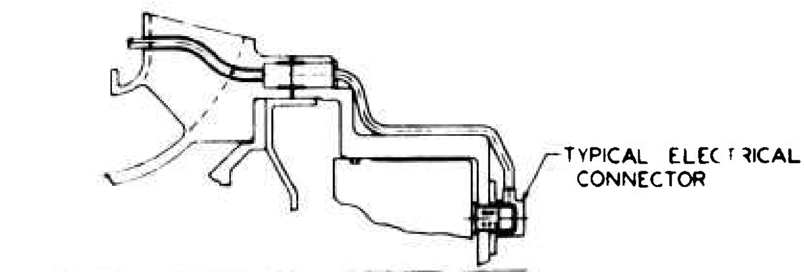
Figure 26. Accessory Gearbox, Nose Configuration, Reduced Weight Version.



SECTION G-G







SECTION C-C

B

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

1. It is concluded that it is not feasible to eliminate all engine mounted accessory drive gearing for engines in the 2-to-5-pound-per-second range. In the 1968-1969 time frame it is not likely that fluid pumps and rotating electrical equipment can be made available which will operate at engine rotor speeds.

   However, a substantial reduction in the size and weight of accessories and accessory drive gearing can be made by the use of higher accessory speeds than those currently used.

2. Because of the speed limitation indicated above, it is concluded that the complete integration of accessories into the basic gas generator is not feasible. The need for a speed reduction requires an accessory drive gearbox that can best be kept external to the gas generator. Any internal arrangement of accessories at this time would complicate maintenance and overhaul procedures.

   It does appear that many benefits can be derived by the integration of the accessories and gear train into the gas generator structure. In this manner serviceability is retained while a modular concept can be implemented.

3. No unique concepts were revealed for power extraction or addition. It was concluded that the most practical system for starting the engine and for extracting secondary power is of the hydraulic type. By using current equipment it was found that there is a three-to-one weight advantage of the hydraulic system over the electrical system. The volume advantage is likewise in favor of the hydraulic system. The results of this study indicate that significant breakthroughs are not likely to change this comparison.

   It is also concluded that the hydraulic starting system, when provided with a manually operated pump, is very attractive from the standpoints of self-sufficiency and suitability for operation from austere sites.

4. A common control system suitable for use on the 2-to-5-pound-per second engines in the 1968-1969 time frame cannot be identified in detail at this time. The fluidic control proposed by Honeywell is, in concept, a candidate for such an application. The general development work being conducted by another control manufacturer* in the area of hybrid electronic/hydromechanical control is also applicable to this requirement.

   It is felt that a specific definition of a suitable common control system can be obtained only after an extensive study is made of the expected engine characteristics and the performance requirements as dictated by potential applications.

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*Bendix Aerospace Division

It can be concluded that commonality can be attained in control subsystem elements, such as speed sensors and temperature sensors, by continued development in these areas. The availability of a highly responsive turbine inlet temperature sensor will contribute significantly to the feasibility of a truly universal engine control.

5. It is felt that the 1968-1969 time frame demonstrator should make use of the following accessories and accessory arrangement:

   a. High speed fluid pumps operating in the 15,000-to-20,000-rpm range.

   b. High speed electrical ignition source that does not require energy storage devices. This unit should operate in the 30,000-to-40,000-rpm range and should provide, as secondary functions, a speed signal and a useful amount of electrical power.

   c. A fluidic speed sensor operating in a speed range of 30,000 to 40,000 rpm to provide an input to the control system if a pneumatic speed sense is required.

   d. Hydraulic motor/pump for starting and hydraulic secondary power extraction. This unit should embody moderate state-of-the-art advances.

   e. All of the accessories noted above should be integrated into a compact arrangement with the necessary gear reduction. The accessory package should contain advanced ideas relative to fluid connections and mounting.

RECOMMENDATIONS

1. It is recommended that the maximum reduction in the weight and size of engine mounted controls and accessories be achieved through the development of rotating accessories that will operate at two to three times their current speeds. This will not only make for a substantial reduction in the size and weight of the accessories themselves, but will permit the use of the smallest possible accessory gearbox. High speed, small units also lend themselves to efficient packaging arrangements. In the case of fluid pumps, the emphasis should be placed on the application of the technology that has been established for missile systems. This would involve reducing the maximum operating speed to 15,000 to 20,000 rpm and extending the life as required. Gear and vane type pumps should be considered in such a development program.

   In the case of electrical generating equipment, it is recommended that only that amount of electrical power necessary to permit self-sufficient operation of the gas generator be extracted from the

gas generator. This will permit the use of high speed, special purpose devices that can be accommodated in an accessory package of minimim size. For instance, it should be possible to combine the functions of ignition power, battery charging and speed sensing in one high speed electrical device.

2. It is recommended that development emphasis be placed upon the efficient packaging of high speed rotating components and accessory drive gearing. The resultant package should be removable as a single unit by unskilled personnel using simple tools. All fluid connections should be foolproof in nature. The accessory package should have a standard mounting arrangement and should be interchangeable with various engine sizes and types within the 2-to-5-pound-per-second range. The accessory package should contain, as a minimum, the fuel and lubrication pumps, the ignition power supply and the fuel control speed sensor. Provisions should also be made for a standard starter attachment. The design should permit a simple gear change to accommodate the various engine speed ranges.

   This concept would take advantage of the results of the component development programs recommended above.

3. It is recommended that development effort be devoted to the perfection of a standardized hydraulic starter system for small gas turbine engines. This effort should include further reduction in the weight and size of the components, an increase in motor/pump operating speed and the development of a compact, lightweight and efficient hand pump. Consideration should also be given to the development of lightweight accumulators in the smaller sizes.

   The hydraulic starting system should incorporate a motor/pump which can provide hydraulic secondary power once the engine has started. The motor/pump and necessary conversion devices should be integrated into the accessory package discussed above.

4. It is recommended that additional work be carried out with the engine control manufacturers to investigate in detail the problems associated with the development of a common, or universal, control for small gas turbine engines. In order to do this it would be necessary to define the expected characteristics of the engine that are to evolve from the USAAVLABS engine component development program and to identify the performance requirements.

   Definitions of compressor characteristics and engine acceleration requirements are particularly important in establishing the feasibility of a simple universal engine control. For instance, the thrust response requirements associated with V/STOL applications imply a high degree of control sophistication and matching of engine and control characteristics.

Continued development of high temperature sensors will contribute greatly to the attainment of a universal engine control.

5. It is recommended that a development program be established to cover accessories of the type that will be required for the 1968-1969 time frame demonstrator. This program would cover the extension of the speed range and durability of small fluid pumps, the development of advanced ignition systems and an effort to attain the full potential of hydraulic starting and secondary power extraction. This program should be paralleled by a similar effort to cover the integration of the advanced accessories into the accessory drive package. Consideration should be given to standardization and the commonality aspects of the integration.

   Sufficient component development and bench testing should be carried out prior to the use of the components on the demonstrator to preclude test delays.

APPENDIX I

PROBLEM STATEMENT - ADVANCEMENT OF SMALL GAS TURBINE ENGINE ACCESSORY TECHNOLOGY

INTRODUCTION

Twenty years of research in the field of gas turbine engines has advanced the technology to a point where VTOL aircraft are a reality. In the area of engine mounted accessories, the state of the art for all practical purposes has remained static except for minor advancements in component size, weight, etc. (i.e., A.N.D. reference chart, aircraft engine accessory drives - approved 30 April 1947). The lack of advancement in the accessory area does not reflect itself too clearly in large engines. The current Army components program general objectives are to reduce the specific weight of 2-to-5-pound-per-second gas generators by approximately 50 percent with corresponding decreases in SFC and installed volume. If current accessories are installed on the advanced gas generators, the weight of the accessory system will exceed the gas generator weight for the 2-pound-per-second machine. The primary objective of the program is advancement in the state of the art of small engine accessory systems with a resultant increase in reliability, reduced unit cost, and decrease in weight and volume.

STATEMENT OF WORK

It is the purpose of this program to solicit conceptual designs and ideas directed toward the advancement of controls and accessories technology as related to gas generators in the 2-to-5-pound-per-second size.

The overall program objectives are as follows:

1. Determine feasibility of eliminating all engine mounted accessory drive train reduction gears for 2-to-5-pound-per-second gas turbine engines.

2. Determine feasibility of integrating various accessories into the basic gas generator.

3. Investigate and define advanced methods of energy extraction and addition.

4. Define a common control system for engines of the 2-to-5-pound-per-second size.

5. Define an advanced accessory system for incorporation and test on a 1968-1969 time frame demonstrator engine (2-to-5-pound-per-second).

The research shall include, but not necessarily be limited to, the following areas:

1. Fluid Pumps
2. Generators

3. Starters
4. Speed Sensing Devices
5. Alternators
6. Ignition System
7. Temperature Indicating System
8. Variable Geometry Devices
9. Engine Controls

The investigation shall include the above items relative to small turbo-shaft engines of the 2-to-5-pound-per-second airflow class with the over-all objective of a large degree of commonality for all engines of the aforementioned class. Conceptual or preliminary design of individual components shall be provided, and preliminary design of an accessory system for 2-to-5-pound-per-second gas generators shall be conducted utilizing individual component design data.

It is not required that the "conceptual designs" related to the above be complete in every detail or be fully evaluated in the hardware form. However, consideration must be given to the practical aspects of the over-all problem, such as environment and duty cycle, and to those aspects associated with the ultimate application.

In presenting proposals covering any of the items listed above, the vendor should include at least the following information:

1. Technical description and discussion
2. Supporting technical evidence (analysis, test results, etc.)
3. Any anticipated limitations relative to environment or operating life
4. Estimates of development effort required to establish the feasibility of the concept (additional design effort and breadboard evaluations, etc.)
5. Impact of successful development of the concept on

    a. Reliability
    b. Cost
    c. Weight
    d. Volume

The responsibility for the installation or incorporation of the various accessories into the engine design will rest with Wright Aeronautical Division. However, the vendors are encouraged to combine and integrate components where possible. Coordination will be carried out between WAD and the vendors to obtain the optimum arrangement of accessories.

Space envelopes for the various accessories are not available at this stage. Needless to say, the emphasis is on minimum size and weight. Wherever possible, functions should be combined to make the best possible use of the available space.

A vendor shoul ^ot hesitate to propose a concept that makes use of a portion of the engine structure as a functional part of the accessory. In such a case the proposal should indicate the space requirements and any special requirements relative to material, environment, lubrication, cooling air, etc.

.he schedule for the program described herein is included as Figure 27. Since WAD is bound by contract to this schedule, any inputs from vendors must conform in order to be given consideration in the WAD final report.

Vendor's proposals for items covered by this problem statement should be in a reproducible form so that they may be included in WAD's final report in total, if desired.

'ETAIL INFORMATION

1. Small Gas Turbine Engine Characteristics

The more significant characteristics of small gas turbine engines in the 2-to-5-pound-per-second size are shown in Figure 28. In general, these characteristics were obtained by taking a WAD engine design and extrapolating as required to cover the 2-to-5-pound-per-second range. The WAD engine design is defined as follows:

W_a = 4.483 pounds per second
TIT = 2200^oF
N_d = 50,000 rpm
HP = 721
SFC = .366
P_r = 8.0:1

While the characteristics shown are not strictly those of specific engines, it is felt that they reflect, in terms of the controls and accessories, the requirements of advanced gas turbine engines of the type under consideration.

Curve (a) indicates the relationship between design N_1 rotor speed, turbine inlet temperature, design horsepower, N_1 spool polar moment of inertia, and design airflow. Curve (b) indicates the requirements for light-off, starter cutout and idle speed in terms of N_1 rotor speed.

Curve (c) reflects the application of a typical compressor characteristic curve to an 8:1 compressor. Curve (d) reflects the fuel flow requirements shown in curve (e) with the following allowances in terms of pressure drop:

Fuel Control Metering Head	150 psi
Filter and Line Losses	50 psi
Duplex Nozzle Pressure Drop	15 psi @ min flow, 170 psi @ max flow
Pump Wear Allowance	60 psi

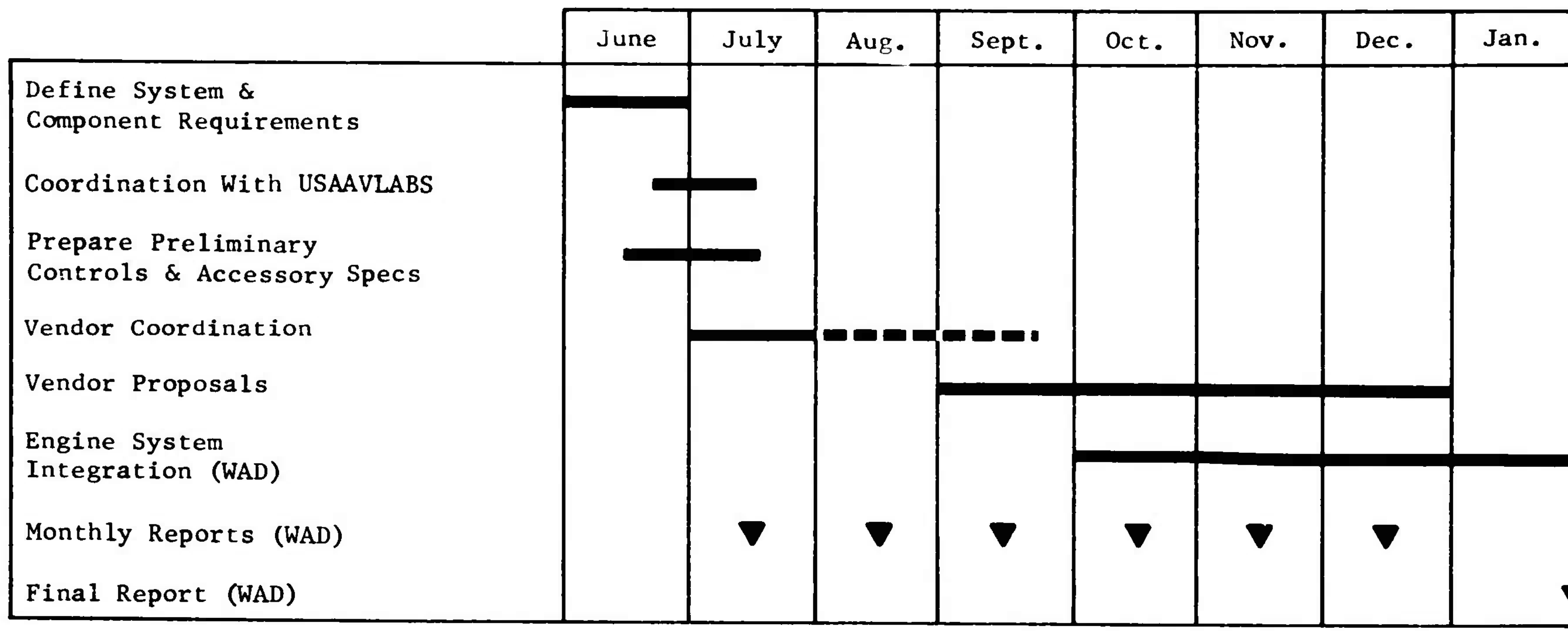
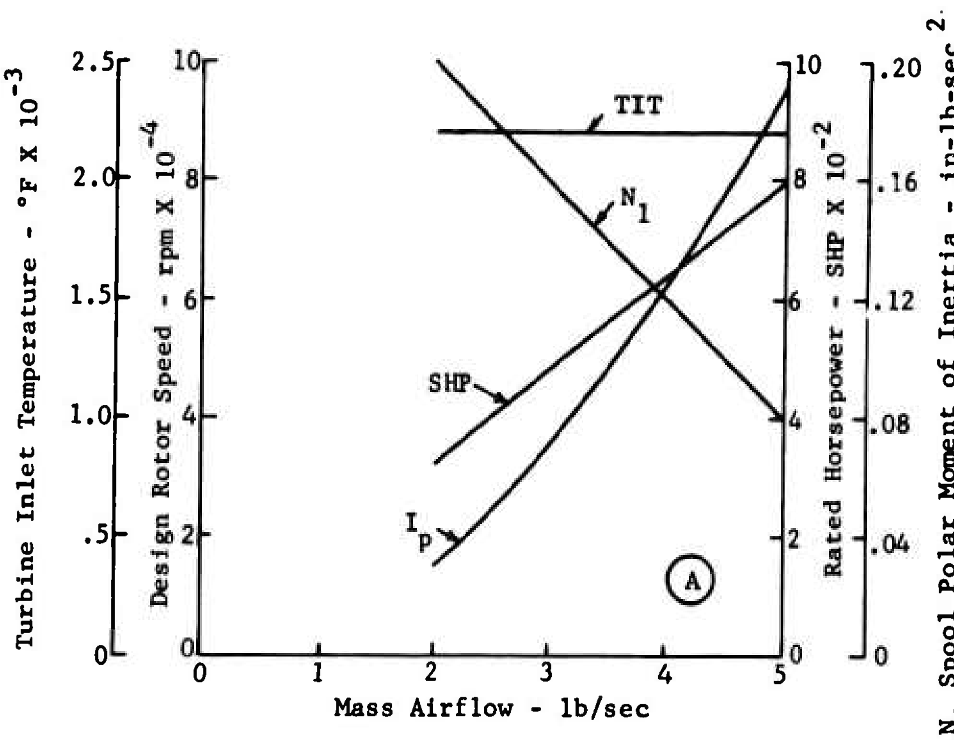
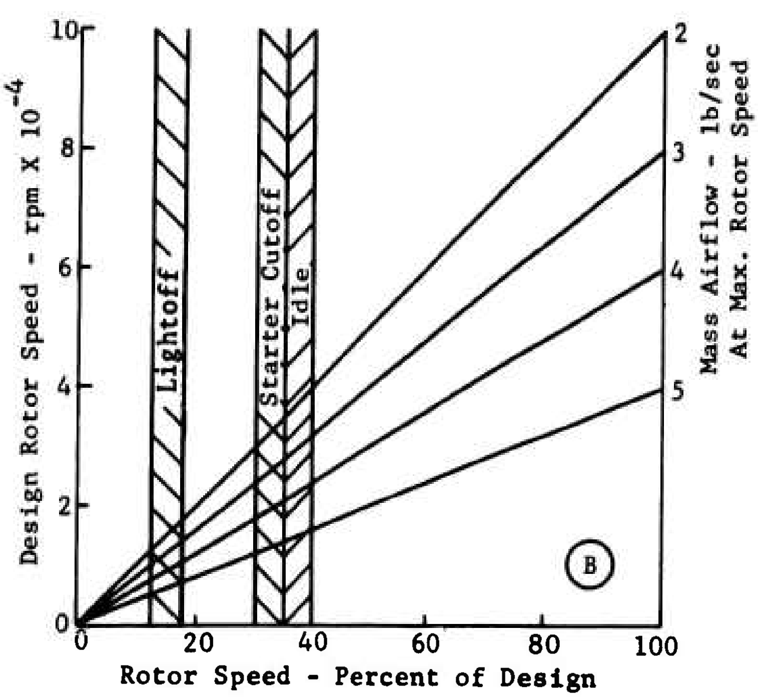


Figure 27. Advancement Of Small Gas Turbine Engine Accessory Technology - Program Schedule.





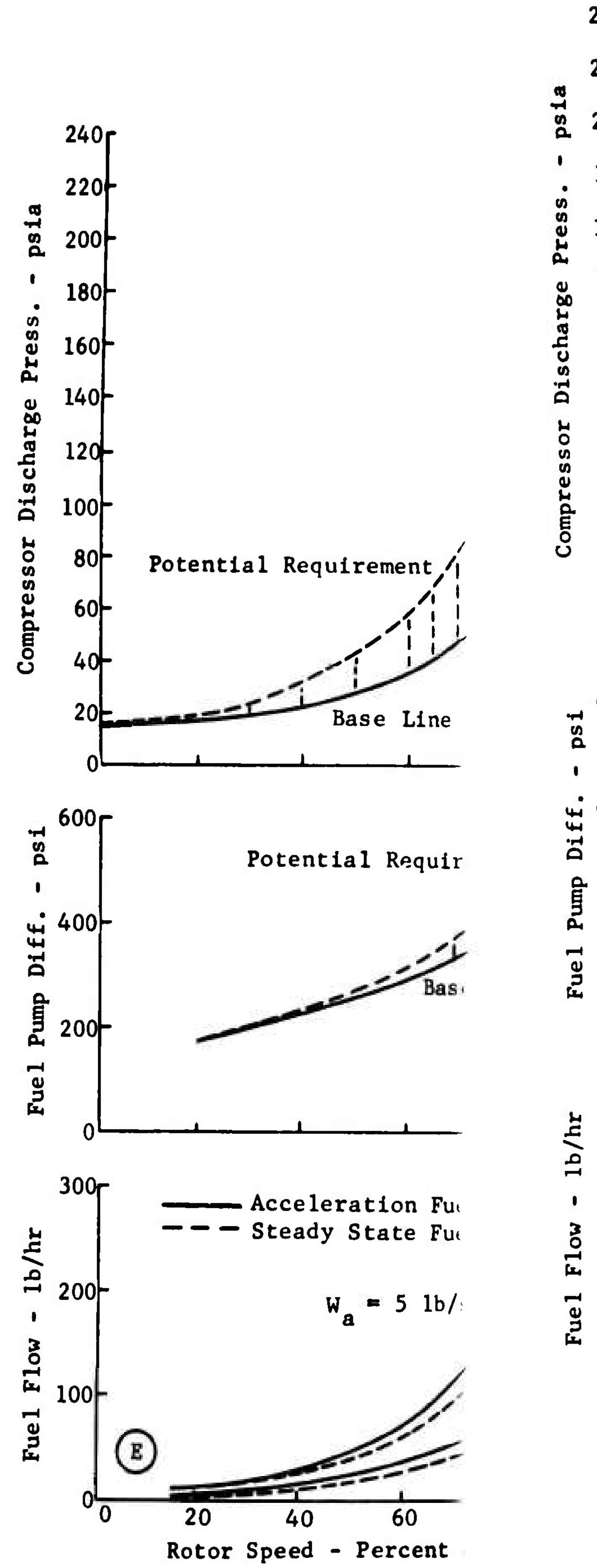
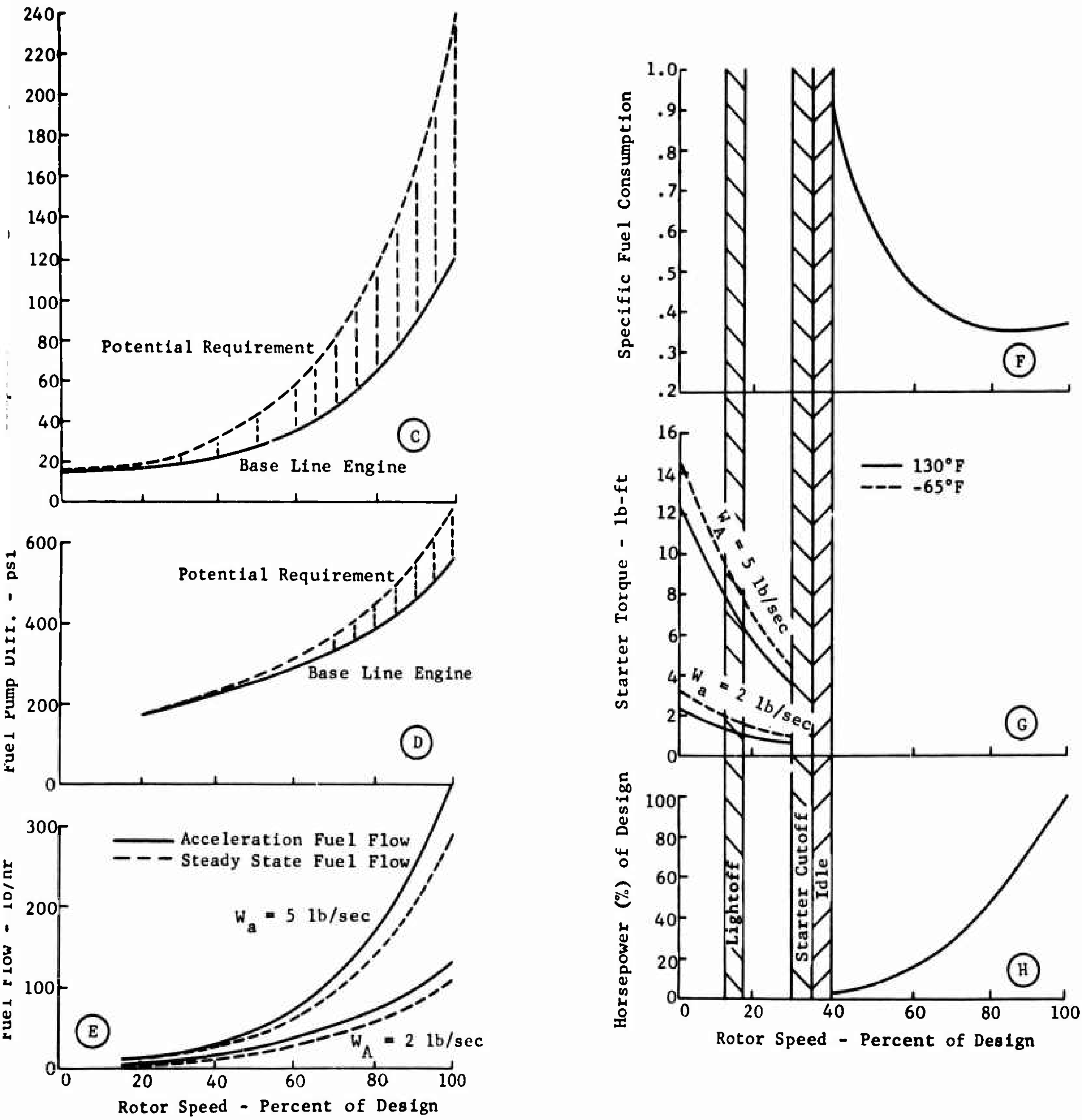


Figure 28. Small Gas Generator Characteristics.


Potential Requirement
Base Line Engine
C
Potential Requirement
Base Line Engine
D
Acceleration Fuel Flow
Steady State Fuel Flow
W_a = 5 lb/sec
W_A = 2 lb/sec
E
Rotor Speed - Percent of Design
Specific Fuel Consumption
F
Starter Torque - lb-ft
130°F
-65°F
W_A = 5 lb/sec
W_a = 2 lb/sec
G
Horsepower (%) of Design
Lightoff
Starter Cutoff
Idle
H
Rotor Speed - Percent of Design

B

A duplex nozzle was assumed to minimize the system pressure required to cover the relatively large fuel flow range.

Curve (e) indicates the steady-state fuel flow requirements as derived from the horsepower curve (a) and the typical SFC characteristic of curve (f). A 20-percent increment was added to the steady-state fuel flow to provide for engine acceleration.

Curve (g) shows an estimate of the starter requirements for engines in the 2-to-5-pound-per-second size. The curves shown are essentially equal to 1 percent of rated horsepower on a hot day and 1.25 percent of rated horsepower on a cold day. These values were selected after a review of all small engine data that are currently available.

Curve (h) indicates the estimated typical relationship between horsepower and N_1 Rotor speed.

2. Fluid Pumps

a. Engine Fuel Pump

The fuel flow and pressure rise requirements for the engine fuel pump are shown in Figure 29. Adequate performance in the areas of lightoff and idle is particularly important, and sufficient allowance should be made in the design of the pump to account for deterioration in performance with operating time.

A definitive specification for the level of contamination of the fuel to be handled is not available at the present time. However, an indication should be provided by the vendor to indicate the degree of contaminant resistance claimed for any given pump concept or design.

As indicated elsewhere, it is desirable to operate the fuel pump at gas generator speed, N_1, in order to eliminate the necessity of accessory drive reduction gearing.

Integration of the fuel pump and other pumping requirements indicated below into a cartridge type arrangement should be considered.

b. Engine Lubrication Pumps

It is recognized that the exact requirements for the engine lubrication system are closely related to the bearing and bearing-cavity design. Therefore, it is assumed that the following general requirements are adequate to identify the overall problem.

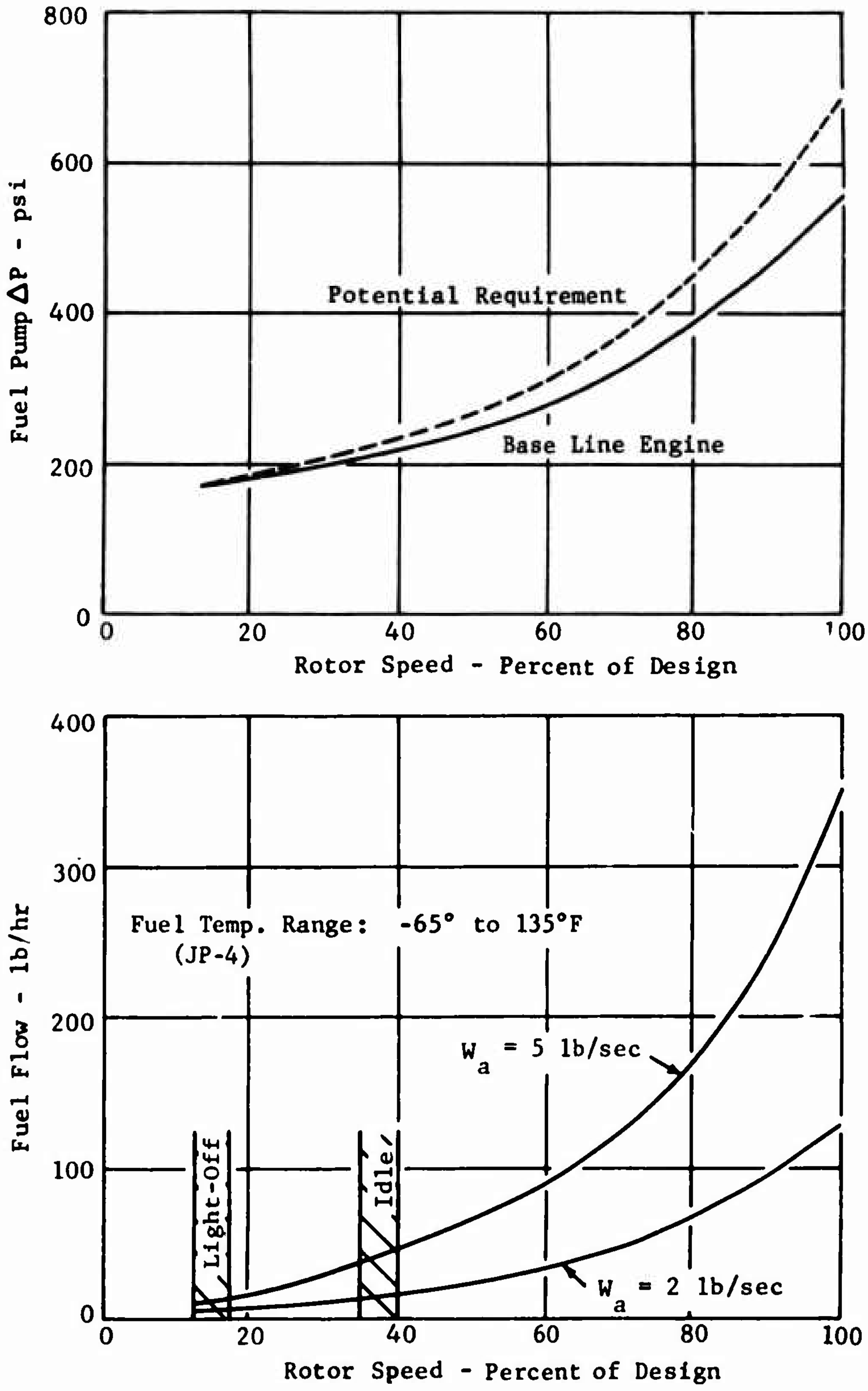


Figure 29. Fuel Pump Requirement.

Lube Pressure Pump

Capacity	5 lb/min at rated rotor speed
Pressure Rise	70 psi
Oil In	250°F
Fluid Type	MIL-L-7808

Scavenge Pump

Capacity	10 lb/min at rated rotor speed
Pressure Rise	30 psi
Oil In	400°F

"Rated Rotor Speed" is intended to cover the range from 40,000 rpm to 100,000 rpm.

NOTE: The above flow rates cover only the requirements for the gas generator. It is assumed that the power turbine and reduction gear would have a separate lube system.

c. Hydraulic Pump

The engine driven hydraulic pump requirements are shown in Figure 30. The intent is to provide accessory fluid horsepower at idle speed equal to 1 percent of engine rated horsepower. System pressure is not specified in order to permit complete freedom in the selection of the approach to be used.

The vendor is encouraged to investigate the use of the hydraulic pump as a fluid motor for starting. Such an investigation should include the sizing of the complete hydraulic system in order to permit complete evaluation of the concept.

3. Generators

The objective here is to provide electrical accessory power equal to 1 percent of rated engine horsepower while the gas generator is at idle speed. From Figure 28 the following values can be obtained:

2 lb/sec W_a engine 3.0 hp @ 35,000 - 40,000 rpm (max of 6.0 hp)

5 lb/sec W_a engine 8.0 hp @ 14,000 - 16,000 rpm (max of 16.0 hp)

The generator output shall be at a nominal 24-volt dc. Again, it is desirable that the generator be driven at gas generator rotor speed to eliminate the need for reduction gearing. Performance characteristics for the generator are to be presented for operation over the complete speed range. Such aspects as electrical noise and radio interference characteristics should be discussed.

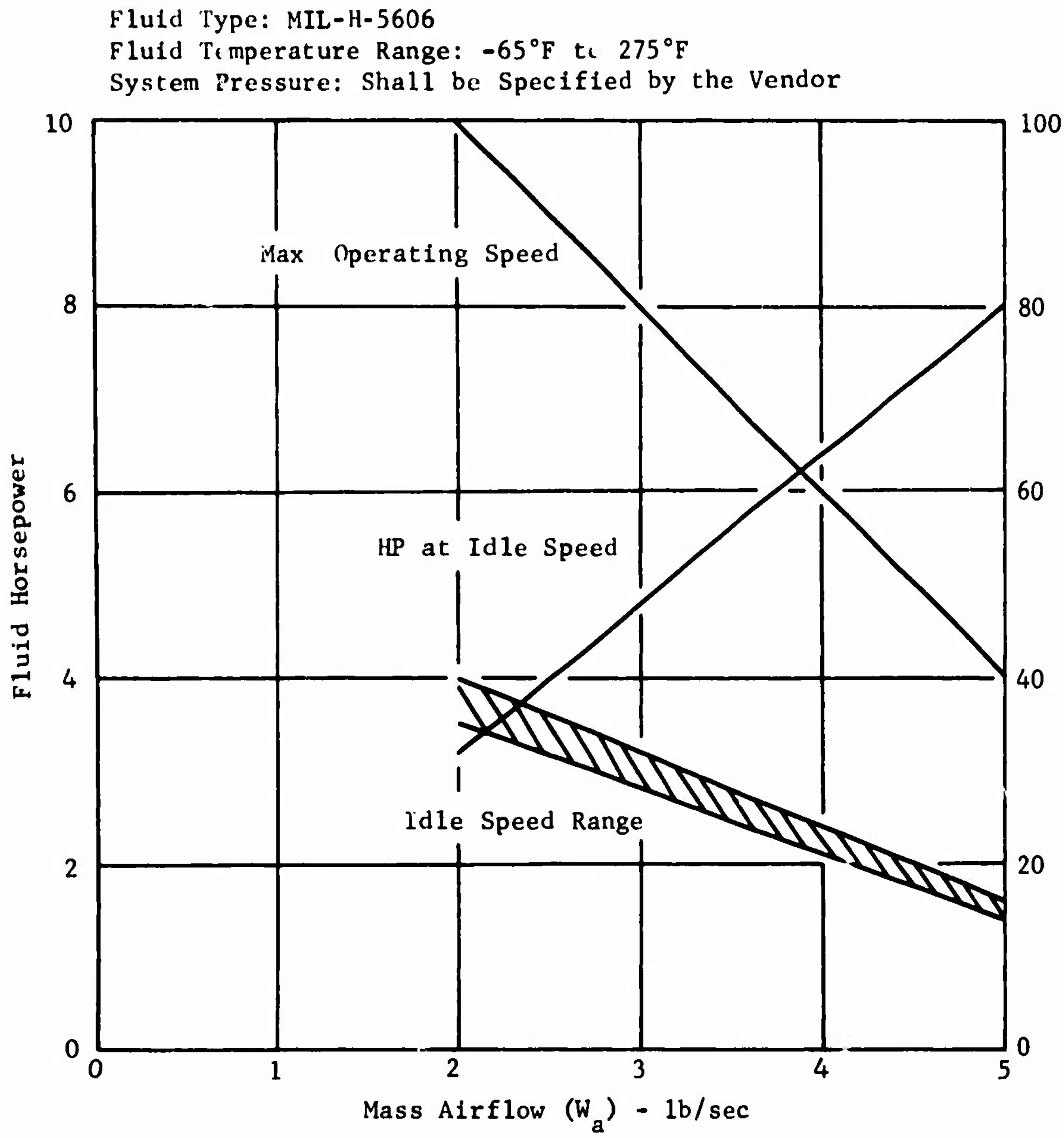


Figure 30. Hydraulic Pump Requirements.

4. Starters

It is highly desirable that the gas turbine engines that evolve from current engine component development work be as self-sufficient as possible. Conventional electrical starters are not considered satisfactory due to the large battery packs or ground power packs required to obtain cold-day starts. New concepts are required to provide self-sufficient means for starting the engine.

The curves of Figure 31 indicate the starter torque requirements as a function of engine size and rotor speed. It is desirable that the starter means be used also to provide accessory power once the engine has been started in order to make use of the dual-function concept. The accessory power requirements are also indicated in Figure 31.

The starter system must permit three starts or start attempts without having to resort to auxiliary energy sources, charging of accumulators, recharging batteries, etc. A typical start will require 20 seconds to cutoff.

Proposals for conceptual designs to meet the above requirements shall cover the complete starter system and shall define all auxiliary equipment. Consideration shall be given to the logistics aspect of the problem and to the use of the proposed system at austere sites.

5. Speed Sensing Devices

In keeping with the objective of eliminating accessory drive gearing and external pads, it is desired that a unique or novel approach be disclosed for sensing and indicating engine speed.

It is not a requirement that the speed sensor be electrical in nature. However, it is required that the sensor output be suitable for use in providing a cockpit indication of engine speed. The proposal shall contain a description of the speed sensor, transducer (if required), and the readout device.

The speed sensing system shall provide accurate and reliable speed indication over the range of rotor speeds indicated in Figure 28; i.e. 12,000 to 40,000 rpm for the 5-pound-per-second engine and 30,000 to 100,000 rpm for the 2-pound-per-second engine. The objective accuracy is $\pm$ 1.0 percent of full scale.

6. Alternators

The objective is to provide electrical power for accessory use from a high speed alternator. The basic power requirements are:

2 lb/sec W_a engine 3.0 hp @ 35,000 - 40,000 rpm (max of 6.0 hp)

5 lb/sec W_a engine 8.0 hp @ 14,000 - 16,000 rpm (max of 16.0 hp)

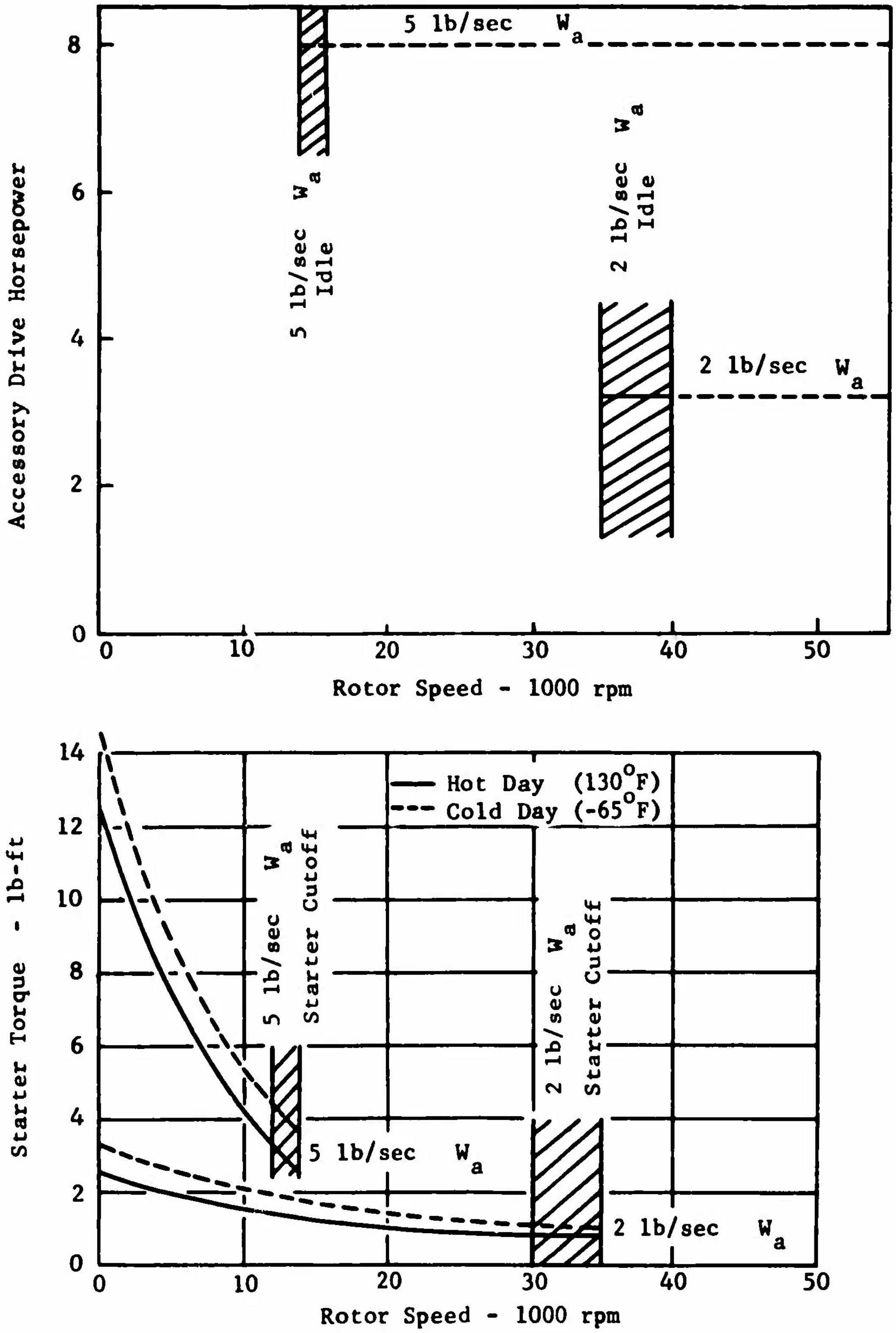


Figure 31. Starter and Accessory Drive Requirements.

The alternator must operate over the complete range of engine speed indicated in Figure 28. The alternator output should be 115 v ac - 400 cps.

Proposals for such a device should include a discussion of the complete system and should cover such aspects as electrical noise, wave forms and radio interference characteristics.

7. Ignition System

The ignition system for the 2-to-5-pound-per-second gas generators must have at least the ignition potential of a two-joule system firing at the rate of two sparks per second. Systems that are completely self-sufficient are desirable. Systems that do not require the conventional energy storage device or exciter unit are also desirable. Proposals for the ignition system must include a definition of the energy source, the igniter plug, electrical leads and any other required equipment.

8. Temperature Indicating System

Two aspects of the small gas generators under consideration make for problems relative to the measurement of turbine inlet temperature. First, the tendency is toward higher temperatures to achieve a high level of engine performance. Second, the gas flow passages are so small that temperature probes of a conventional nature constitute excessive blockage.

The basic requirement is for a temperature sensing and indicating system capable of providing accurate temperature measurement up to temperatures of 2200°F to 2500°F in flow passages of approximately 1/2 inch in height. Local hot spots may approach 3000°F.

The temperature sensor need not provide an electrical output in accordance with thermocouple practice. However, the temperature sensor in conjunction with appropriate transducers and readout device must provide a suitable cockpit indication of turbine inlet temperature.

Proposals for a temperature indicating system for the above requirement should contain a complete discussion of the various elements of the system and should include consideration of multiple point sensing or averaging.

9. Variable Geometry Devices

Small gas turbines of the type under consideration may have variable compressor or turbine geometry. New approaches utilizing hydraulic, pneumatic or electromechanical actuation techniques are required to implement small, lightweight systems.

The following characteristics are considered typical of those that might be required:

Actuating Force	300 lb
Stroke	± 0.5 in. (linear or rotary equivalent)
Rate	3 in/sec
Positional Accuracy	± 0.2 percent full stroke
Input Signal:	
a. Mechanical Position	± 0.25 in. full stroke
b. Pressure	5 to 30 psi full stroke

It can be assumed that mechanical feedback may be used.

Proposals for systems meeting the above requirements must identify the type and level of auxiliary power required to service the actuation system.

10. Engine Controls

The engine control under consideration must have a high degree of flexibility and be suitable for use on various engines of the 2-to-5-pound-per-second class with a minimum of modification. It is not necessary that the investigation be confined to hydromechanical type controls. Pneumatic, electronic, and hybrid systems should be considered.

The engine control must provide the following functions:

a. Starting fuel flow schedules
b. Safe and rapid acceleration and deceleration
c. Full-range governing of N_1 speed
d. Adjustments for idle and maximum speed
e. Overtemperature limiting
f. Free turbine overspeed limiting

Consideration shall also be given to the adjustments necessary to accommodate various grades of fuel and to the incorporation of a manual or emergency mode of operation.

The control concept must feature a high degree of reliability and reflect the requirement to operate with contaminated fuels.

All pressure, temperature, and speed sensors used as part of the engine control system must be adequately defined.

The engine defined below is to be used for the control study.

Design W_a = 4.353 lb/sec
Design N_1 = 54,000 rpm
Design P_r = 9.5:1

Polar Moment of Inertia of N_1 Spool = 0.146 in-lb-sec^2

Operating Range: Sea level static with consideration of capability of operation up to 300 knots at 20,000-foot altitude

Environment: That associated with subsonic aircraft or ground application

The engine compressor is defined by the map of Figure 32. The compressor turbine characteristics are shown on Figures 33 and 34, and the power turbine characteristics are shown on Figures 35, 36 and 37.

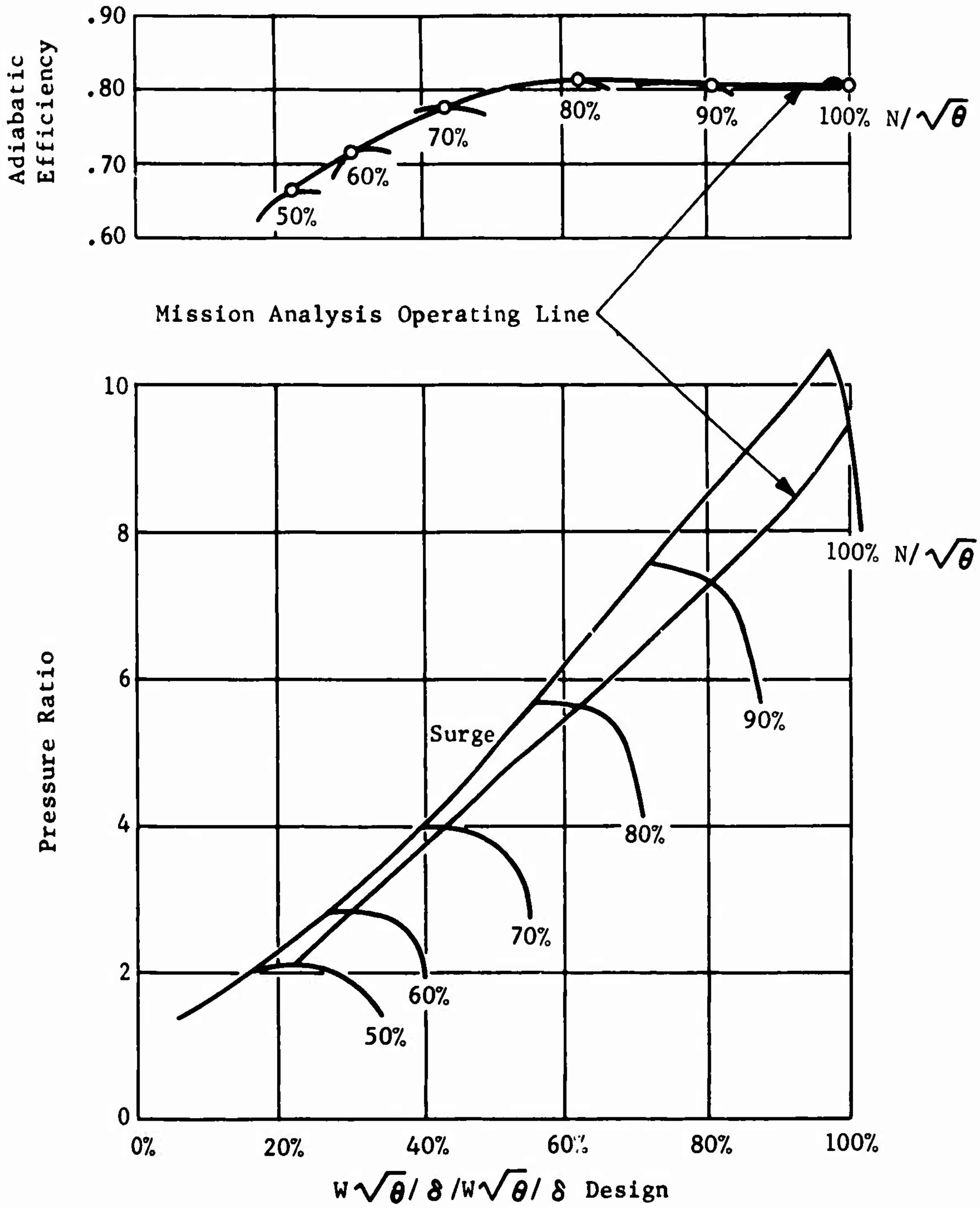


Figure 32. Free Turbine Engine 9.5:1 Pressure Ratio - Variable Power Turbine Stators Axial - Centrifugal Compressor.

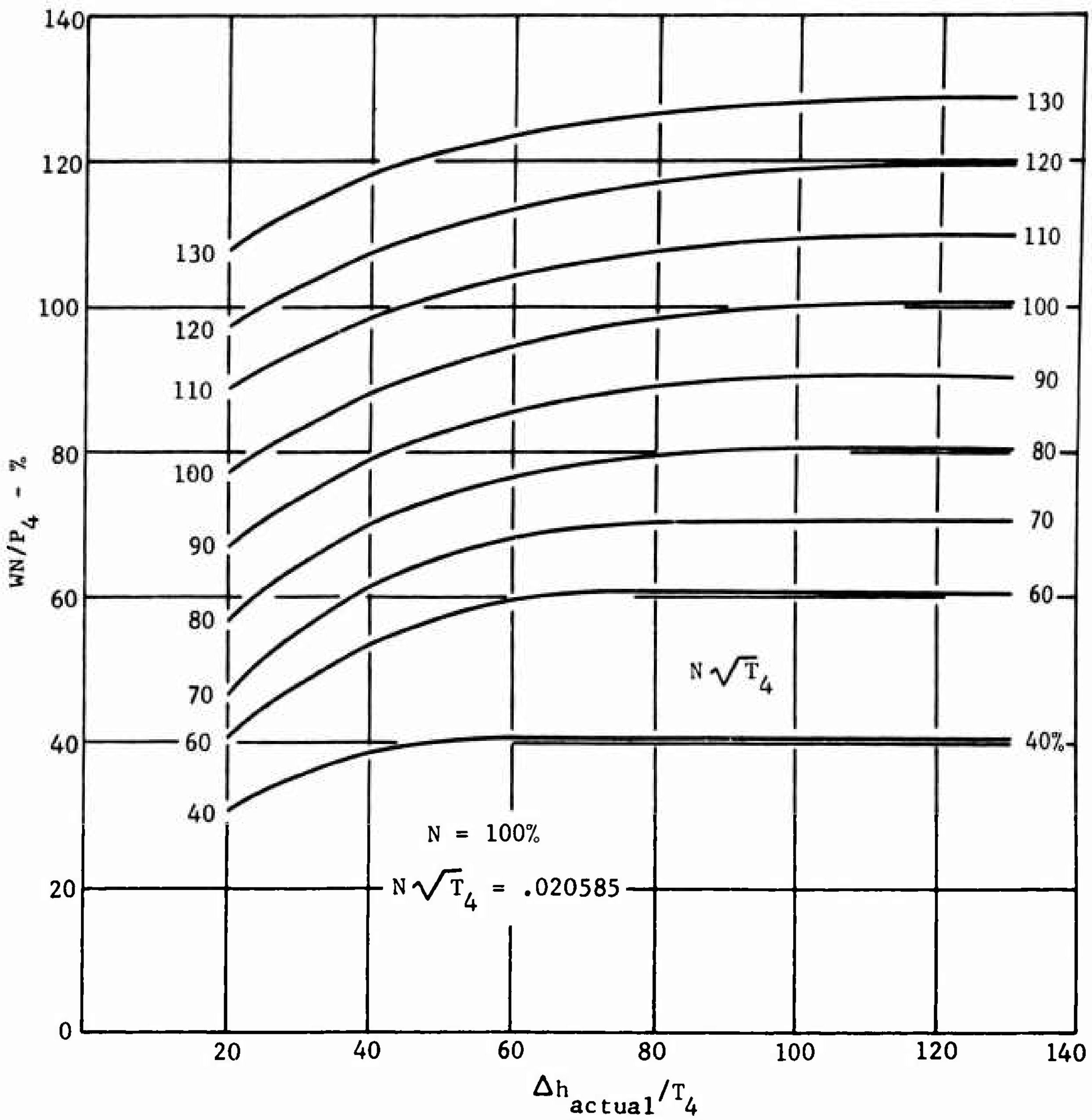


Figure 33. Compressor Turbine Characteristics.

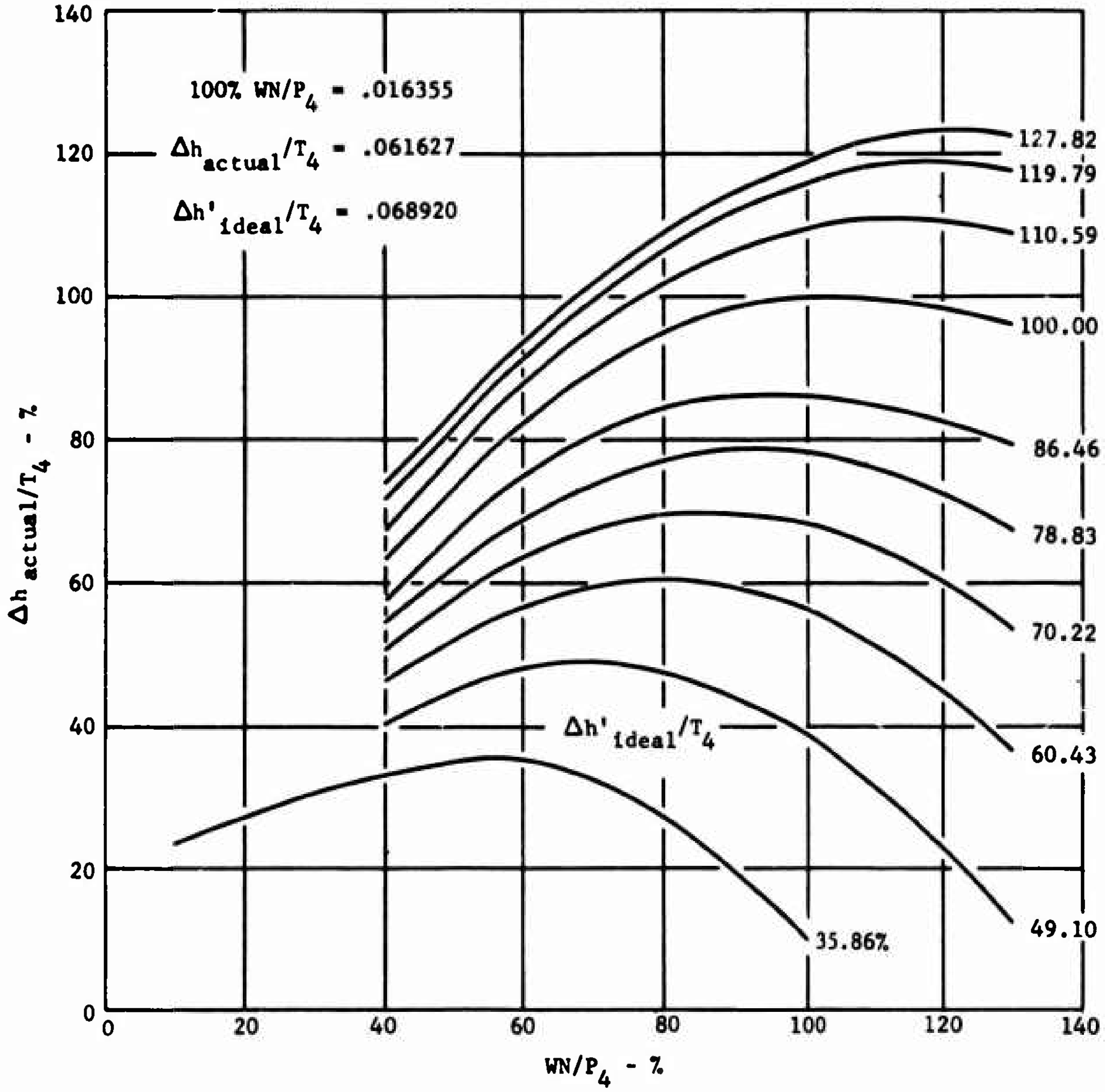


Figure 34. Compressor Turbine Characteristics.

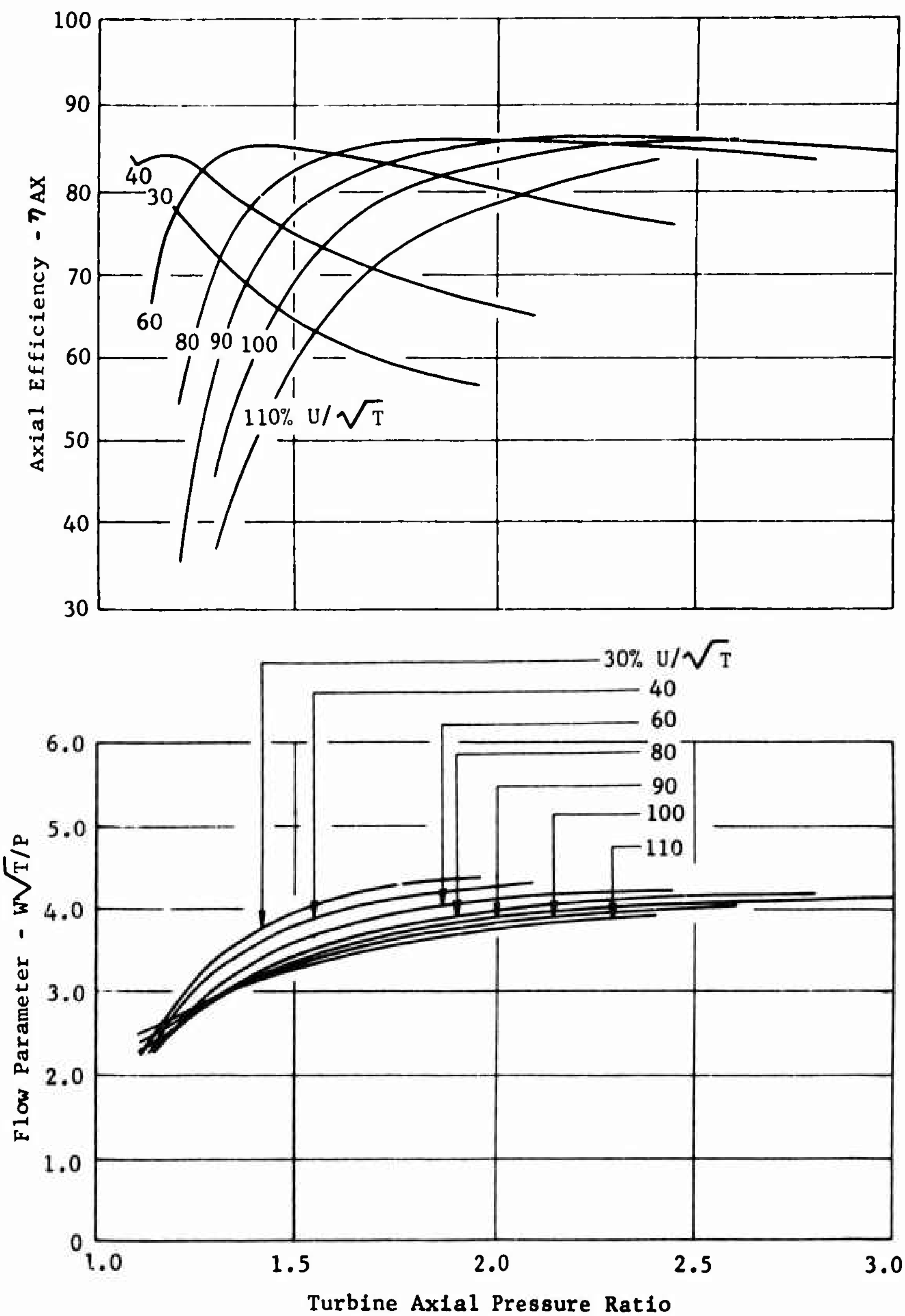


Figure 35. Two-Stage Power Turbine Performance Map - 100% Nozzle Area (-5° Design Incidence Angle - Both Rotors).

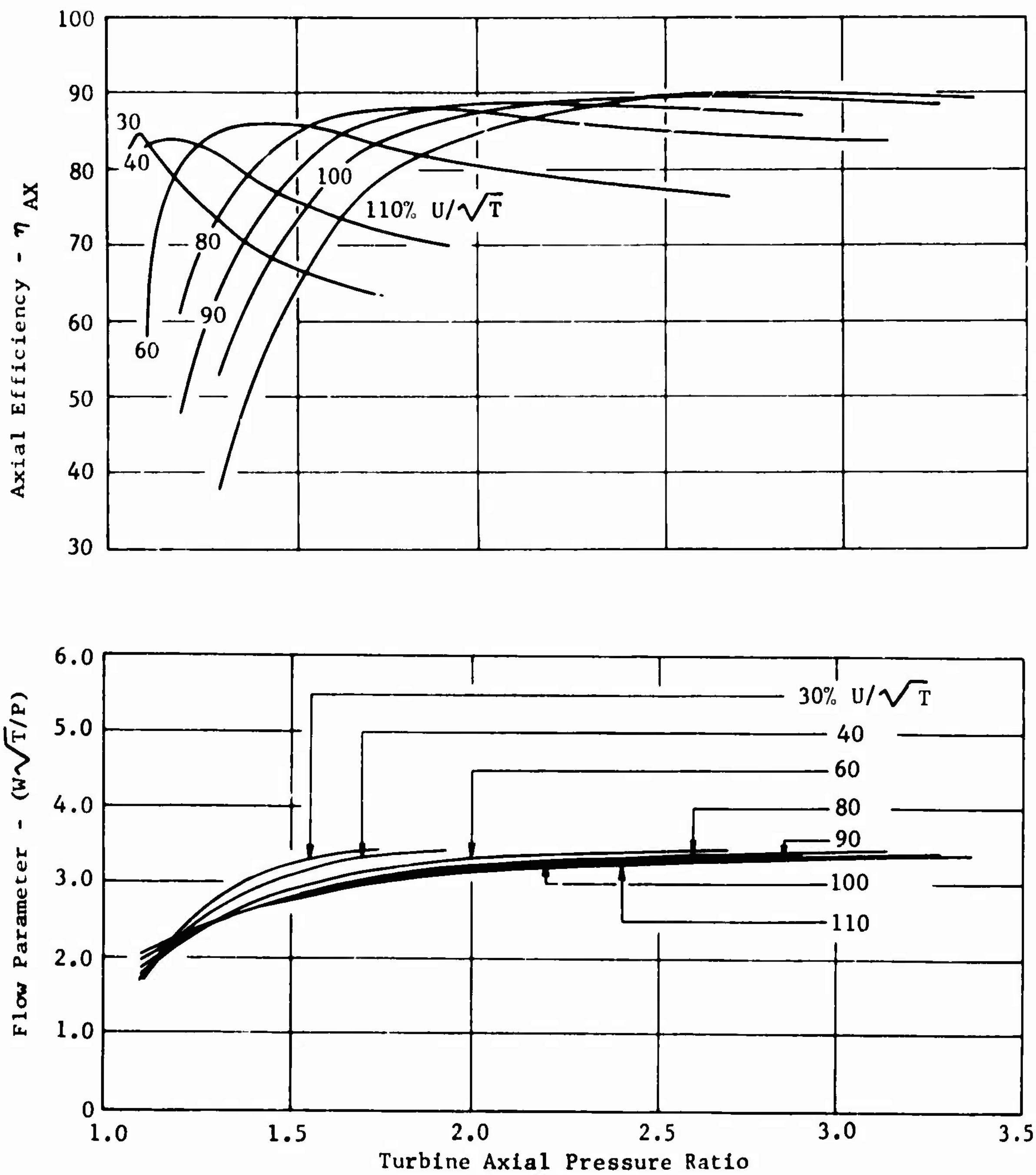


Figure 36. Two-Stage Power Turbine Performance Map - 75% Nozzle Area (-5° Design Incidence Angle - Both Rotors).

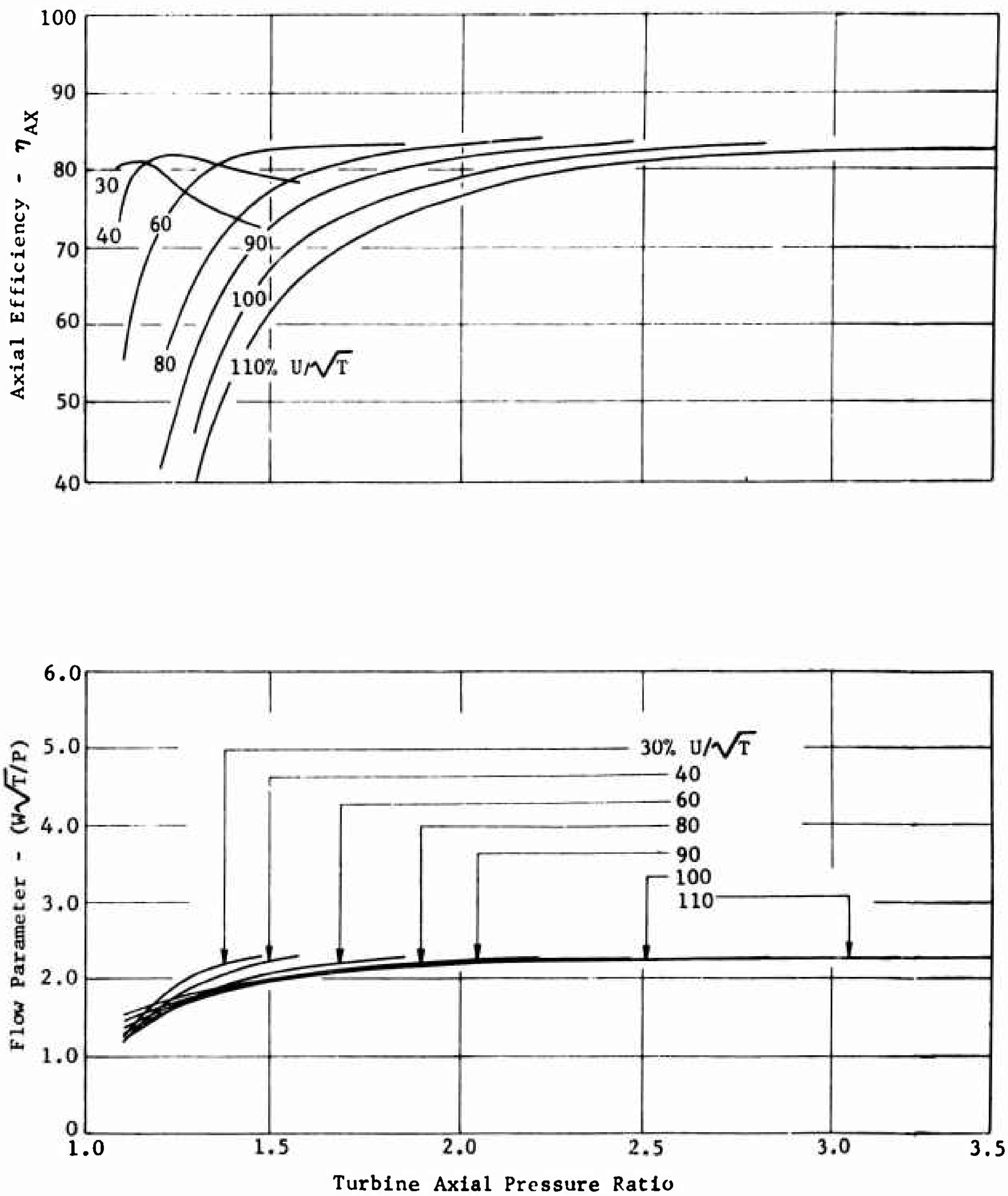


Figure 37. Two-Stage Power Turbine Performance Map - 50% Nozzle Area (-5° Design Incidence Angle - Both Rotors).

APPENDIX II

LIST OF ACCESSORY AND CONTROL MANUFACTURERS AT WRIGHT AERONAUTICAL DIVISION VENDOR BRIEFING

The following companies were represented at the open-house vendors' briefing for the USAAVLABS/WAD Advanced Accessories Study Program held at WAD on July 21, 1965:

USAAVLABS Personnel

Wright Aeronautical Division Personnel

Accessory Manufacturers Represented

A.M. Meeting	P.M. Meeting
Bendix Research	Chandler Evans Inc.
Honeywell	Woodward Governor Company
Garrett Corporation	Bendix Utica
Sundstrand Corporation	Bendix Scintilla
Bendix Aerospace	Vickers Inc.
Fairchild-Hiller (Stratos)	Hamilton Standard
General Laboratory Associates	ADEL Div./Trans America
Power Equip. Div/Lear Siegler	TRW/Accessories Division
Bendix - Red Bank	Borg-Warner
General Electric Company - Accessories	

The following companies were contacted and provided with Problem Statements subsequent to the briefing held at the Wright Aeronautical Division:

Giannini Controls Corporation
Astromechanics Research Division
179 Lancaster Avenue
Malvern, Pennsylvania

Delavan Manufacturing Company
West Des Moines, Iowa

Huyck Systems Company
Division of Huyck Corporation
Huntington, New York

Science Products
U.S. Route 46
Dover, New Jersey

Hurletron-Whittier Incorporated
750 W. Rivera Road
Whittier, California

Consolidated Controls Corporation
15 Durant Avenue
Bethel, Connecticut 06801

Stewart-Warner Corporation
1514 Drover Street
Indianapolis, Indiana 46207

Rocketdyne Division
North American Aviation, Inc.

Pratt and Whitney Aircraft
East Hartford, Connecticut

Tech Development, Incorporated
2601 Timber Lane
Dayton, Ohio 45414

Kidde Aero-Space Division
Walter Kidde & Company Inc.
Belleville 9, New Jersey

National Water Lift Company
400 Delancy Street
Newark 5, New Jersey

Inter-Controls Incorporated
7904 Old Georgetown Road
Washington, D.C.

Mechanical Technology Inc.
Latham, New York

Holly Carburetor Company
11955 East Nine Mile Road
Warren, Michigan

Roper Hydraulics, Inc.
P.O. Box 630
West Caldwell, New Jersey

Barber-Colman Company
Rockford, Illinois

R.E. Phelon Company
East Longmeadow, Mass. 01028

Vickers Incorporated
Great Neck, L.I., New York

APPENDIX III

SMALL ENGINE CONTROL STUDY

BACKGROUND

One of the principal limitations to further improvements in thrust-to-weight ratio of small gas turbine engines is due to the current design approach employing individually packaged engine-mounted accessories and associated accessory drive reduction gears. If accessories of current design were installed on advanced gas generators, the accessory system weight would exceed the gas generator weight for a 2-pound-per-second W_a machine.* General objective of the current Army components program is to reduce the specific weight of 2-to-5-pound-per-second gas generator systems by approximately 50 percent and to decrease specific fuel consumption and installed volume. Increase in reliability and reduced unit cost are other primary goals.

CONTROL SYSTEM OBJECTIVES

The purpose of this phase of the study is to investigate the feasibility of developing an engine control system which meets these basic objectives:

1. Requires no engine-mounted accessory drive train reduction gears.
2. Provides a high degree of flexibility with minimum modification required for various gas turbine engines in the 2-to-5-pound-per-second W_a class.
3. Incorporates engine speed and temperature signals for pilot instrumentation which do not require accessory gear drives.

The control system is to provide the following functions:

1. Starting fuel flow scheduling
2. Safe and rapid acceleration and deceleration
3. Full-range governing of gas generator speed
4. Adjustments for idle and maximum speed
5. Overtemperature limiting
6. Free turbine overspeed limiting

Consideration also will be given to adjustments needed to accommodate various grades of fuel and to incorporation of a manual mode of operation.

*Curtiss-Wright, "Problem Statement-Advancement of Small Gas Turbine Engine Accessory Technology", WAD S-161, July 7, 1965.

STUDY ENGINE SPECIFICATION

The subject of study is a small "paper" engine for which the principal design point data were supplied by Curtiss-Wright. It is a free-coupled turboshaft engine with the following characteristics at the design point:

$W_a = 4.353$ lb/sec

$N = 54{,}000$ rpm

$P_{3_T}/P_{2_T} = 9.5$

$I = 0.01217$ ft-lb-sec^2

$T_B = 0.020$ sec

Altitude: Sea level static; consider also operation up to 20,000 ft and 300 knots

Environment: Subsonic aircraft or ground application

INFLUENCE OF CONTROL APPROACH ON ENGINE PERFORMANCE

TODAY'S PRESCHEDULED CONTROL

The life of an engine is strongly dependent on the time spent at peak temperature. Today's gas turbine engines are controlled through a hydromechanical system of diaphragms, valves, levers, and cams which preschedule fuel flow according to several measured operational parameters. The slow response of temperature sensing elements that are compatible with the hydromechanical control preclude a closed-loop control about the most critical operating parameter, turbine inlet temperature. The fuel flow schedule must for safety cut back fuel sooner than actually necessary. The margin of safety built into a hydromechanical control must be wide enough to protect even a worn and dirty engine with high operational time from surge, or overtemperature operation.

ACHIEVING HIGHER PERFORMANCE

While overtemperature operation must be avoided in order to achieve long life, if an engine is to yield high efficiency it must operate close to its maximum permissible temperature. Direct, rapid sensing of turbine inlet temperature (T_{4T}) has long been sought for achieving closed-loop temperature limiting of gas turbine engines. The importance of this requirement is underscored by the fact that future high performance engines will achieve their higher yield by operating at significantly higher temperatures. An increase of 400°F over present peak temperatures would double the thrust from engines of the same weight.

ADVANCED CONTROL SYSTEM

Under Air Force contract, Honeywell is engaged in development of an integrated propulsion control whereby air withdrawn from the engine is used to power a fluidic system of sensors and amplifiers fitted to the engine for control of starting, stopping, acceleration, deceleration and relight. The fluidic sensors operate with no moving parts and are capable of performing in adverse environments heretofore impossible or impractical on operational gas turbine engines. Since pneumatic control systems can measure parameters which are more indicative of engine operation and can be completely integrated with multiengine VTOL flight control systems (and inlet controls), a simpler and more effective control system without prescheduled fuel flow is possible. In consideration of these advantages and since the fluid system can be adapted to an engine without accessory drive reduction gears, it was considered a good candidate for this requirement.

TEMPERATURE RESPONSE

Fluid systems are built to have dynamic response characteristics required of present-day turbojet engine controllers. In some instances the frequency response of the fluid component is considerably better than the equivalent electrical or hydraulic counterpart. For example, the Honeywell turbine inlet temperature sensor has a time constant less than one-tenth that of a compensated thermocouple used at the turbine inlet. Compensation of a thermocouple is usually fixed, and since the thermocouple time constant varies with engine speed, the effect is to provide overcompensation at one end of the speed range and undercompensation at the other.

COMPUTATION RESPONSE

Because pneumatic signals can only be transmitted at the speed of sound, fluidic signal conditioning and transmitted circuits are slower than electronic circuits. While this is offset by the generally faster response of fluidic sensors, consideration must be given to response of fluidic signal conditioning circuitry in the overall system analysis.

CONTROL APPROACH COMPARISONS - FACTORS OTHER THAN PERFORMANCE

In this section comparisons are made of the chief characteristics of candidate control systems which are unrelated to engine performance, particularly those characteristics related to future growth potential of advanced engines. Conventional pneumatic control systems are not specifically discussed because they are considered to be early state of the art whose ultimate development is the fluid system approach.

RELIABILITY

Reliability predictions summarized in Table III indicate that the fluid engine control systems will be four to five times as reliable as current hydromechanical systems and more than twice as reliable as purely electronic systems. Hydromechanical failure data, reported by one airline for the period January 1964 through May 1964, are compared in Table III with predicted reliability experience for electronic and fluid systems.

Estimation of electronic system performance was derived by applying Honeywell failure rates to electronic components considered necessary to satisfy the functional requirements of an engine control system. The fluid system estimate was made by applying available reliability data on similar components such as gaskets, pneumatic joints, and valves.

The fluid system is inherently more reliable because the component parts are fewer in number, of a simple nature, and relatively immune to adverse environments such as high temperature and vibration.

TABLE III
COMPARISON OF CONTROL SYSTEM RELIABILITIES

Reliability Parameter	Control System: Hydromechanical	Electronic	Fluid
MTBF (Hours)	4,920	8,800	20,000*
Failure Rate (%/1000 Hours)	20.12	11.35	5.00

*This figure applies to the more complex type of integrated control being developed for the U.S.A.F. Simpler controls will have corresponding higher MTBF.

The high inherent reliability of fluid engine controls will enable most system reliability requirements to be satisfied with a nonredundant system. However, where redundancy is required, either for increased reliability or crew safety, the fluid system is again a logical choice.

The small size and light weight of fluid components permit redundant mechanization without the associated size and weight penalties of the hydromechanical system and with less complexity than electronic systems. Flexibility of mechanization is available in that fluid systems are adaptable to partial redundancy limited to valves and actuators (the least reliable components), as well as to complete system redundancy.

DEVELOPMENT FLEXIBILITY

During development, changes are often required in the engine control due to changing system performance or functional requirements. Hydromechanical controls are difficult to modify due to the need to maintain integrity of the high pressure package. Because of the low pressure supply (5 psig) used with fluid systems, fluid stages are as easily reworked as electronic stages.

INSTALLATION FLEXIBILITY

Prescheduled hydromechanical controls are designed specifically for each engine model. Closed-loop controls of the fluidic or hybrid (fluidic sensors and electronic circuits) type are potentially more flexible in application to a variety of engines with a minimum of modification.

In a given class of engines, such as the 2-to-5-pound-per-second W_a category, two component functions of the presently conceived fluidic control system will need to be changed if a given control is to be installed on a different model engine: the surge line computer and the start fuel scheduler. Provision for other necessary adjustments such as the overtemperature and overspeed limits can be designed into the control system.

The subsystem nature of the fluidic implementation makes it a simple matter to provide for interchanging of surge line computers and start fuel schedulers. However, the current Air Force integrated engine control contract includes a task for the development of a surge sensor. Successful conclusion to this task would allow closed-loop surge control. If the approach to surge sensing is economical to implement, the use of turbine inlet temperature for determining the surge line can be eliminated along with the need to substitute different surge line computers when adapting to different engines.

ACCOMMODATION TO VARIOUS FUELS

The closed-loop nature of the fluidic type fuel control makes it ideally suited to burning of various grades of fuel without need for modifications.

It also appears possible to start the engine on various fuels without changing the start fuel scheduler, with only the elapsed time to idle being affected. This is a preliminary judgment which needs to be verified by testing.

COST

Today's control system on a subsonic commercial turbojet accounts for perhaps 30 percent of the propulsion system cost. The simpler control loops and fabrication techniques made possible by fluid technology should permit corresponding economies over the existing system.

In the case of future high performance engines, it is difficult to make direct comparisons because of the probability that fluidic engine controls will be required to handle more functions than are possible with hydromechanical systems. Furthermore, control systems of the future will operate in hotter, noisier environments requiring, in certain cases, sensors constructed of the same advanced materials as the stator blades, often integrated as part of the functional engine components.

WEIGHT

Electronic control systems appear at first to be the lightest weight of all. However, when environmental factors are considered, electronic systems must be penalized for additional weight due to cooling required in hot zones or due to remote location in moderate temperature zones. In addition, sensors located in hot zones will necessarily be of a nonelectronic nature. Because of the electronic cooling provisions and the probable hybrid nature of the ultimate system, a meaningful weight comparison between fluid and electronic approaches is difficult to make at the present time. On an item-by-item comparison, for example, fluid speed sensing would weigh less than an electrical tachometer-generator or pulse generator. However, electronic computing and logic elements might be lighter in weight than corresponding fluid elements but will be more complex.

In comparing systems of similar sophistication, the hydromechanical control is the heaviest of the approaches considered.

FUTURE GROWTH POTENTIAL

The hydromechanical control contributes indirectly to engine weight through dependency on the accessory drive pad. Future engine growth potential will be enhanced if the engine control is capable of operating without a gear-driven accessory pad.

CROSS MODULATION

One possible application for small, high performance gas generators is to provide lift or thrust propulsion for VTOL aircraft. Depending on the cluster configuration of a specific multiengine VTOL, automatic engine control inputs may possibly be furnished only by the attitude control and rate sensing signals. For another configuration it may be necessary, for engine-out protection, to supplement these signals with a limited-authority, cross-modulation control. This might be accomplished by linking the multiple lift engines through their control systems, for example, as a function of compressor discharge pressure.

Cross modulation appears to be much more feasible for fluid or electronic systems than for the hydromechanical control.

RESPONSE

Compared with prescheduled hydromechanical controls of similar sophistication, it is believed that fluid systems potentially are capable of faster response. The fast response of the fluid turbine inlet temperature sensor is particularly advantageous because it permits direct, closed loop control of engine acceleration and also results in a simpler temperature control.

STATE OF THE ART

The singular advantage of the hydromechanical approach is the highly developed state of the art. The use of hydromechanics has probably reached its peak reliability potential with future improvements largely limited to eliminating or simplifying functions. Electronic technology is also highly developed, although the only electronic engine controls currently in service are being used by the British.

The fluidic or hybrid fluidic/electronic approach seems in many respects ideally suited for engine control. Fluid technology is, however, in the development stage.

APPROACH SELECTED FOR STUDY

FLUIDIC CONTROL

Because of the problems associated with achieving higher engine performance and the advanced nature of the integrated fluid control system versus the present prescheduled approach, it was decided to study the feasibility of adapting integrated fluid controls to satisfy the projected dynamic requirements and "padless" operation of the small engine.

The emphasis of the basic fuel control study was on system simplicity commensurate with sea-level-static or low-altitude flight operation. Additional functions needed to provide altitude capability to 20,000 feet were considered briefly. Fluidic methods of automating the engine start and relight sequence were also studied for those applications where these features will be needed.

Consideration was also given to engine speed and temperature signals for pilot instrumentation which do not depend on gear drives. Hybrid transducers using fluidic sensors and electronic circuits were considered for control or display applications.

COMMAND INPUT PARAMETER

The objective of the engine control system is to control the gas generator so that the delivered energy will meet the requirements of the load turbine. Hot gas energy delivered to the load turbine can be measured only indirectly as a function of the temperature, pressure, and mass flow rate. However, the pressure ratio and flow rate of the compressor are essentially functions of rotor speed for steady-state conditions. Therefore, at steady-state speed, either the speed or the pressure ratio can be considered as representative of the energy being delivered to the load turbine.

Because of simpler hardware implementation and ready sources of engine performance data, the "Speed Command" type of control system was chosen for first investigation. Safe engine acceleration is achieved by a closed loop temperature limit schedule control which has been tested on an engine by NACA* and which was adapted for this study to meet the anticipated requirements of the small engine. A CDP command system was selected as a backup approach.**

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* Gerus, Theodore F., et al, A Temperature-Schedule Acceleration Control For A Turbojet Engine and Its Use with A Speed Control, NACA Research Memorandum RM E5 7118a, December 3, 1957.

**A significant advantage projected for the CDP control was that the CDP pressure responds more quickly than rotor speed to changes in fuel flow. This gives a slightly leading signal which would prove useful in event of stability problems due to a short time constant in the speed control loop.

ENGINE SIMULATION

It was thought that the acceleration time constant could be estimated by extrapolation from existing turbojet engines. A plot of contemporary engine data is shown in Figure 38. This figure shows that the time constant is roughly proportional to the mass flow to the one-half power.* On this basis, time constants for engines in the 2-to-5-pound-per-second W_a category would range from 0.05 to 0.17 second, and burner time delay would become a critical factor in controlling the engine. Burner time delay was estimated to be approximately 20 milliseconds by WAD.

The anticipated time constant of the engine was very low. Therefore, Honeywell deemed it necessary to simulate the fuel control/engine combination on a computer to ensure stability of the system using fluid components. The simulation was also needed to check the operation of the simplified version of the integrated engine control, utilizing a proportional-plus-integral speed control loop with only a proportional temperature limiting loop. Initial simulations used an engine time constant of 0.08 second. Final simulations employed a time constant curve determined analytically using calculated data applying specifically to the small engine.

It was decided to simulate the engine and control system on a hybrid (combined analog and digital) computer system. Due to the nonlinear characteristics of the engine over the full range of operation, this method provides a better simulation of the dynamic characteristics of the system than the linearized analog approach.

In order to implement the engine simulation, it was necessary to estimate some of the acceleration characteristics. To save time and cost to the program, an engine map was obtained by combining the design data presented in the WAD problem statement with closely related data from production engines.

*The equation plotted in Figure 38 was developed from an equation originally presented in Appendix D, University of Minnesota M.S. Thesis, 1955, by Ross D. Schmidt.

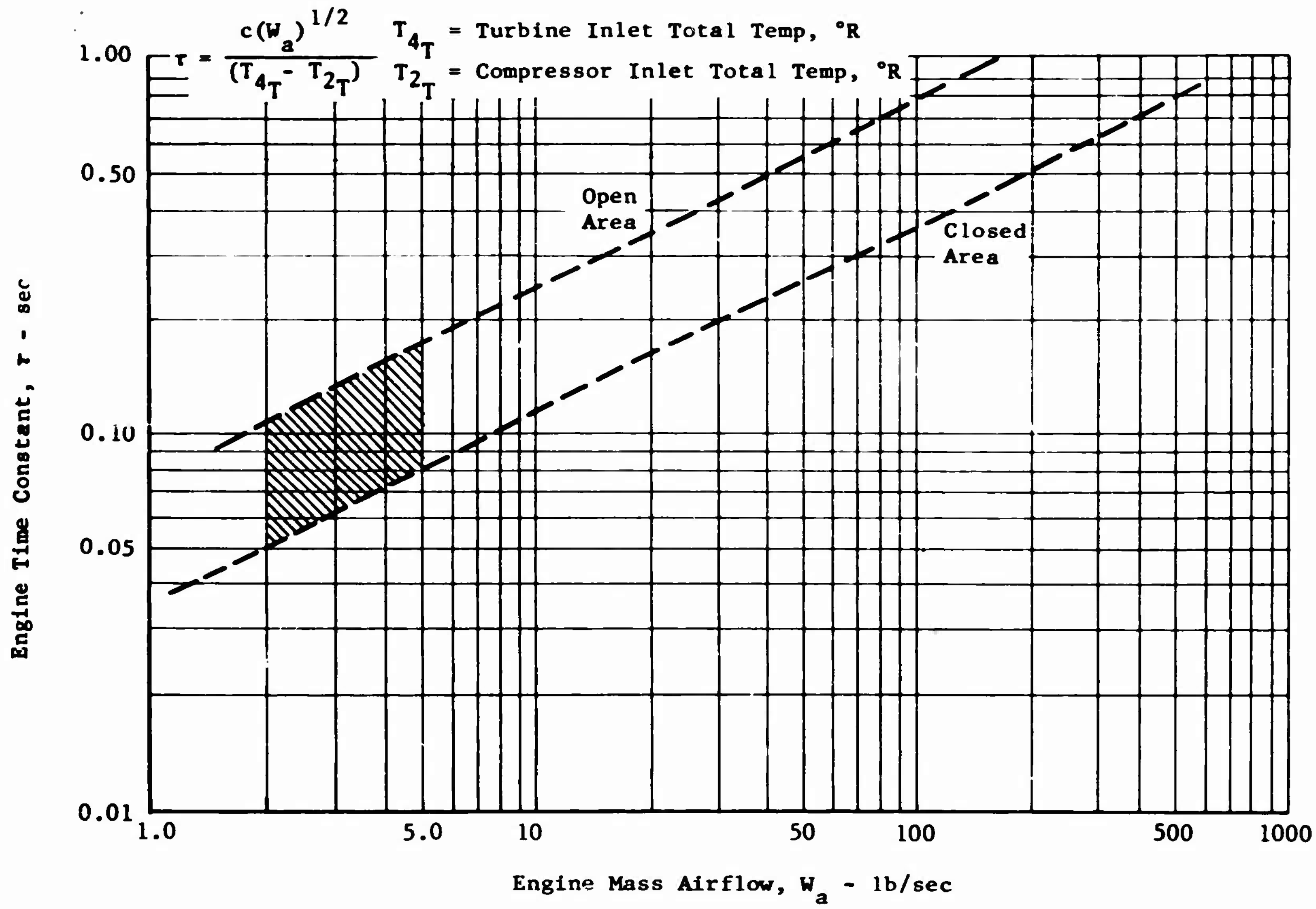


Figure 33. Trend of Engine Time Constant Estimation.

COMPUTED ENGINE DYNAMICS

BACKGROUND

The Wright Aeronautical Division small engine being studied here is a "paper engine" for which the principal design point data have been supplied by Curtiss-Wright. In order to simulate this engine on the hybrid computer, it has been necessary to estimate some of the acceleration characteristics. As explained under ENGINE SIMULATION, some of this has been done by adapting results of earlier work on other engines where applicable.

However, investigation has shown that the acceleration time constant of extremely small engines cannot be determined this way. This is due to scaling effects resulting from greater than normal hub-to-tip diameter ratio and proportionately heavier shafting in a small flow engine. Accordingly, the time constant has been determined analytically using calculated data applying specifically to this engine. A description of the computational procedure and related performance curves are given in Annex A of Appendix III.

CONCLUSIONS

The time constant at any steady-state speed is shown in Figure 39. Results indicate a dynamic response which is typical of engines having 10 times the rated flow of the small engine. It was concluded that the "paper" engine under study presented no unusual problems in the transient response required of the speed control loop. This removed any doubt that the "speed command" type of fuel control was worthy of further study for possible application to the small engine.

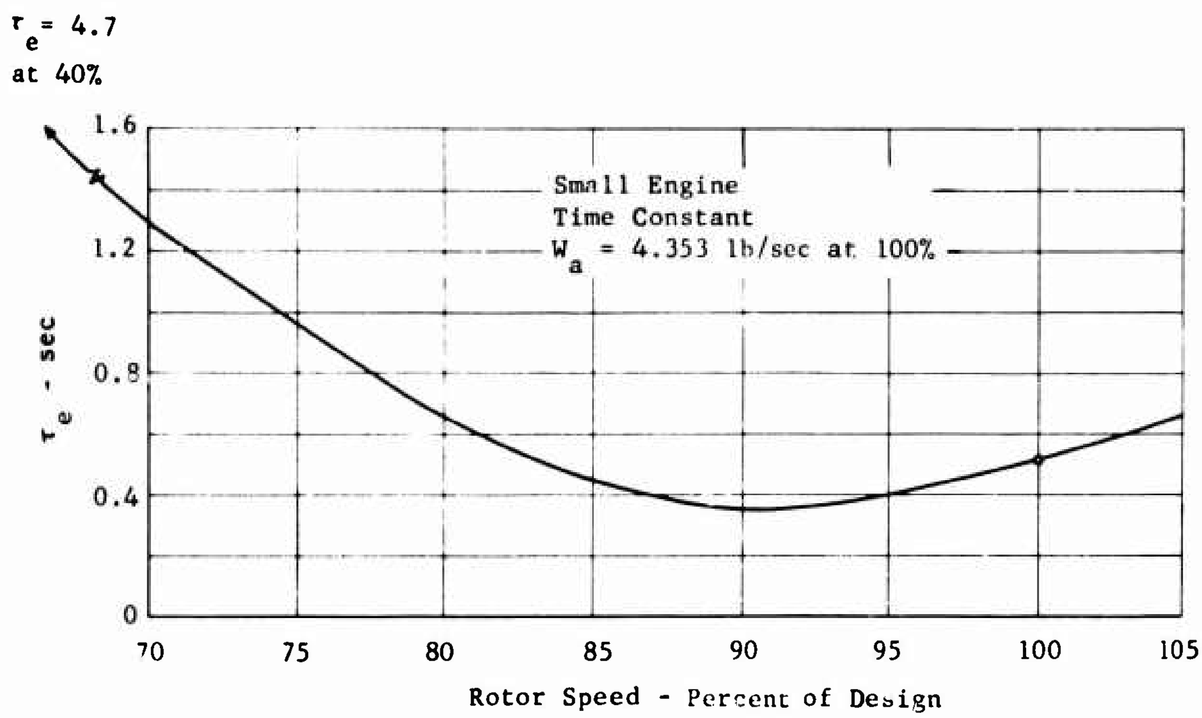


Figure 39. Time Constant Versus Steady-State Speed.

BASIC FUEL CONTROL

SYSTEM DESCRIPTION

The basic fuel control system recommended for the small engine is a proportional-plus-integral speed controller with a turbine-inlet-temperature override system. Turbine inlet temperature is used to ensure that excess engine temperatures are not encountered as well as to set an acceleration limit as the engine comes up to speed.

While this system is basically simple in both concept and mechanization, it has the capability of reasonably rapid acceleration from idle to full speed without entering the surge region. The system is compensated for temperature effects, but because of the limited applicational environment of the small engine, pressure, altitude and Mach number, compensation to obtain constant engine response characteristics is not required. Therefore, the proposed system is much less complicated than the fluid system Honeywell is currently developing for the U.S. Air Force.

The use of turbine inlet temperature for determining the surge line is advantageous for a number of reasons. First, a temperature limit is required in any event, and it is reasonable to use it for a dual purpose. Second, if dynamic surge or stall is inadvertently encountered, the sudden increase in temperature assures a recovery from this condition (there is a hysteresis condition associated with surge and/or stall, and therefore fuel flow must be reduced somewhat below the value which caused the condition). Third, setting the surge limit with turbine inlet temperature is more desirable than scheduling fuel flow because, as the engine ages, the chances of encountering surge become greater with a fuel scheduled system than with a temperature scheduled system. Preliminary indications are that the design margin from the surge boundary does not decrease as drastically, if at all, using the fluid, temperature-scheduled loop. Fourth, the surge line can be computed without the aid of pressure information which is needed for fuel scheduling. This complicates a fuel schedule system and is one of the causes of the relatively low reliability found in hydromechanical systems.

Finally, because the end-effect temperature is used for setting and measuring the acceleration limit, the variable effect of fuel density is bypassed and, therefore, compensation for this effect is not required as it is for conventional controls.

A block diagram of the system is shown in Figure 40. The basic speed control loop is redundant with respect to the two-speed sensors. The temperature loop and separate overspeed sensor serve to restrict the gas generator to safe operating limits. An additional overspeed sensor (not shown) can be added to restrict the speed of the free coupled load turbine. The temperature limit is of dual importance because in high performance engines the line of maximum allowable engine temperature tends to approach the operating line at maximum engine speed. A function generator is included in the temperature loop to generate a set temperature as a function of corrected speed. The temperature limiting control compares measured turbine inlet temperature with the set temperature.

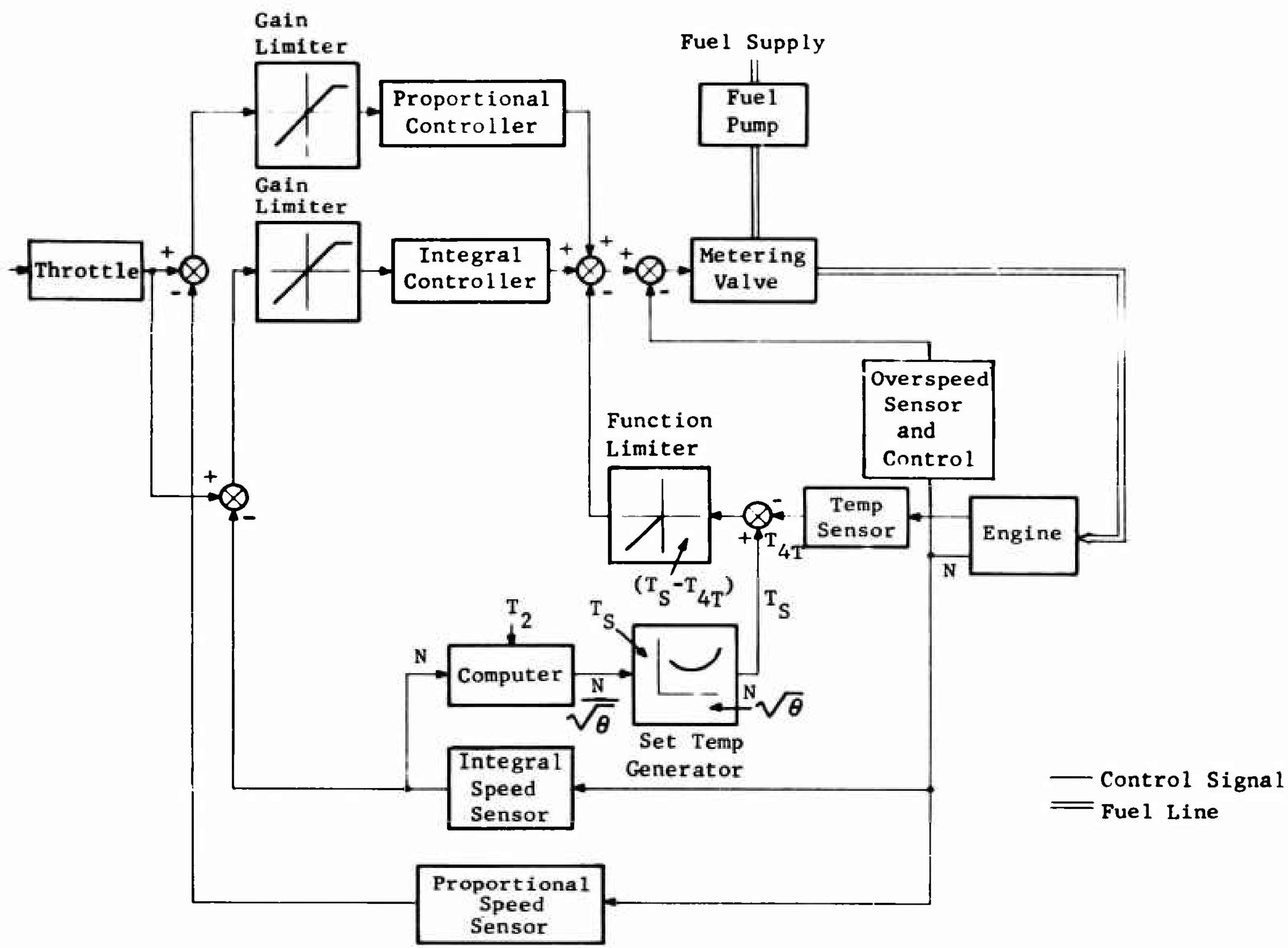


Figure 40. System Block Diagram - ic Fuel Control.

When the measured temperature is greater than the set temperature, the fuel flow is reduced by the control system. As long as the measured temperature remains below the set temperature, the output of this limit type of temperature loop is zero. The purpose of the function limiter shown in the temperature loop is to block positive signals tending to increase fuel flow, passing only negative signals tending to decrease fuel flow.

Although the proportional-plus-integral speed control is stable by itself, addition of the set temperature limiting circuit makes the combined system become unstable. In the interest of simplicity, stabilization was obtained by inserting a gain limiter in each speed loop rather than using a variable gain changer. When the speed error exceeds a preset value, a reduction in gain of the speed loop serves to limit sharply further increases in speed error signal amplitude.

SYSTEM ANALYSIS

Most of the system work was done on the hybrid computer. However, preliminary stability analysis using the root locus approach was used to support the computer results. Although the root locus solutions are valid only for engine operation in the neighborhood of the point about which linearization takes place, they do provide initial insights into the problem as well as a check on the computer simulation.

A stability analysis was conducted on the speed control loop without including the nonlinear gain limiter. The transfer function of the proportional-plus-integral speed loop is shown in Figure 41. The dynamics of the system components shown are typical of present-day hardware. Since the mechanization of the control was not fixed, only components with the most significant lags and delay times were included. The effect of the fluid amplifiers on the stability of this system is small, and therefore, it was omitted for this analysis.

SPEED LOOP ANALYSIS

To obtain the open loop transfer function for plotting the root locus, the form of the proportional-plus-integral speed controller was changed to

$$K_P + \frac{K_I}{s} = K_I \left[\frac{K_P/K_I s + 1}{s}\right]$$

This results in an open loop transfer function of

$$GH = K_I K_M K_N K_S \left[\frac{(K_P/K_I\ s+1)\ e^{-0.04s}}{s(1+0.02s)\ (1+0.01s)\ (1+\tau s)}\right]$$

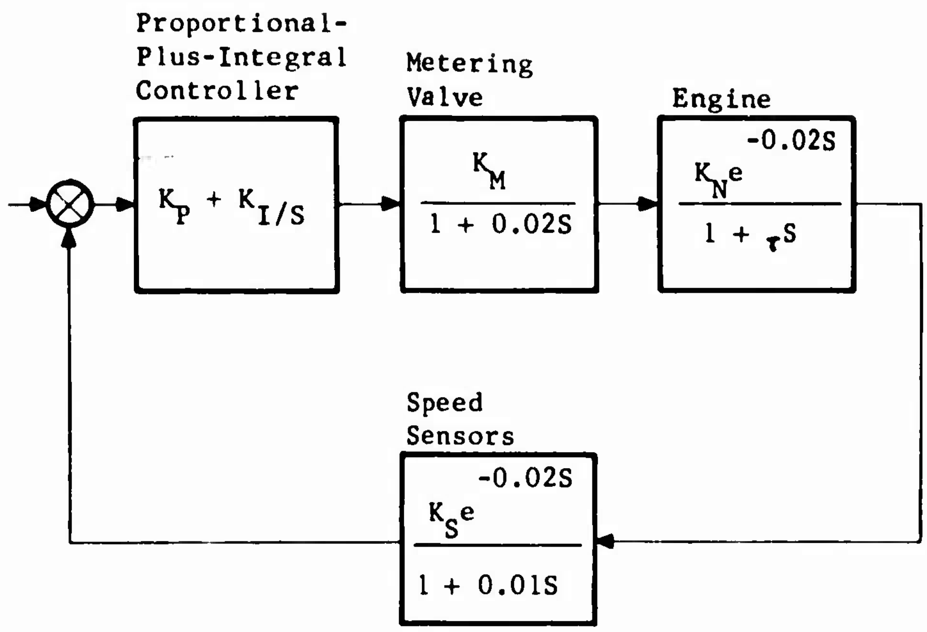


Figure 41. Transfer Function - Speed Control Loop.

From the hybrid simulation the best value of the ratio of proportional-to-integral gain (K_P/K_I) was found to be in the neighborhood of 0.86. The root locus of the speed loop was obtained, from the equation for GH, on the digital computer at several points over the operating range of the engine. For comparison purposes, the open loop gain for a damping ratio of 0.7 was also computed for these cases. The root locus plot for 100-percent speed is shown in Figure 42. Figure 43 gives a comparison between engine gain and the open loop gain required for the speed.

(The open loop gains required to maintain a 0.7 damping ratio decrease by a factor of 10 between 40- and 100-percent speed.) This would at first seem to indicate that gain scheduling as a function of speed would be necessary. However, since the engine gain decreases by a factor of 7 over the same range, gain scheduling as a function of speed may not be required.

Analysis of the speed loop, although certainly not complete, has been carried sufficiently far to indicate that the proposed control system can be used over the entire operating speed range. This is further substantiated by the hybrid simulation. The temperature loop can be analyzed separately in the same manner. However, since component dynamics and configuration of the temperature sensor are classified, a stability analysis of the temperature loop was not included in this unclassified study. The temperature loop was included in the computer simulation which is described in the following section.

COMPUTER SIMULATION

Simulation Approach

Simulation of the engine and basic fuel control was conducted on a hybrid computer with standard day conditions assumed. Hybrid computation uses both analog and digital computers simultaneously to simulate complex systems. Due to the nonlinear characteristics of the engine over the full range of operation, this method provides a much better simulation of the dynamic response of the system than does the analog approach. Nonlinear simulation allows coverage of the entire operating range from below idle to over 100-percent speed. The Honeywell hybrid computer facility used for this study is shown in Figure 44.

One of the most powerful features of the hybrid system is the 21-inch oscilloscope display. This is the primary means of data transmittal, since it gives an instantaneous display of both the engine trajectory and the surge line. This display enables rapid engineering decisions concerning the control system to be made at the console instead of waiting for the normal printer readout.

To implement the engine simulation, an engine performance map was obtained by combining the design data presented in Problem Statement WAD S-161 with closely related data from previously investigated production engines. The engine map was stored in the digital portion of the computer and the analog portion was used for the control system simulation. The digital computer operated on the stored engine map to provide both the dynamic and steady-state engine characteristics.

$\tau = 0.5$

$$GH = K_I K_M K_N K_S \frac{(0.86s + 1)\ e^{-0.04s}}{s\ (1 + 0.02s)\ (1 + 0.01s)\ (1 + 0.5s)}$$

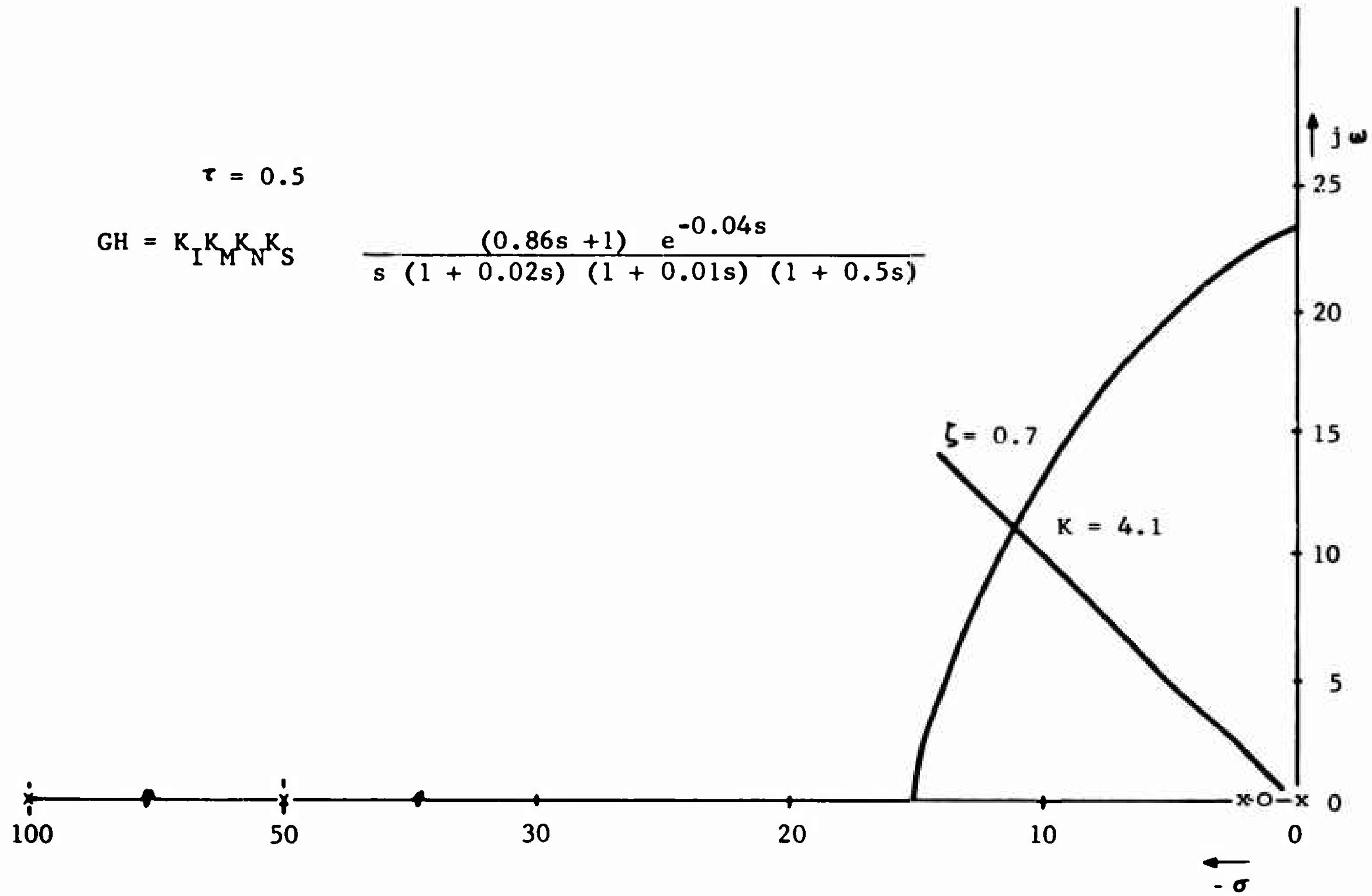


Figure 42. Root Locus of Proportional-Plus-Integral Speed Control at 100-Percent Speed.

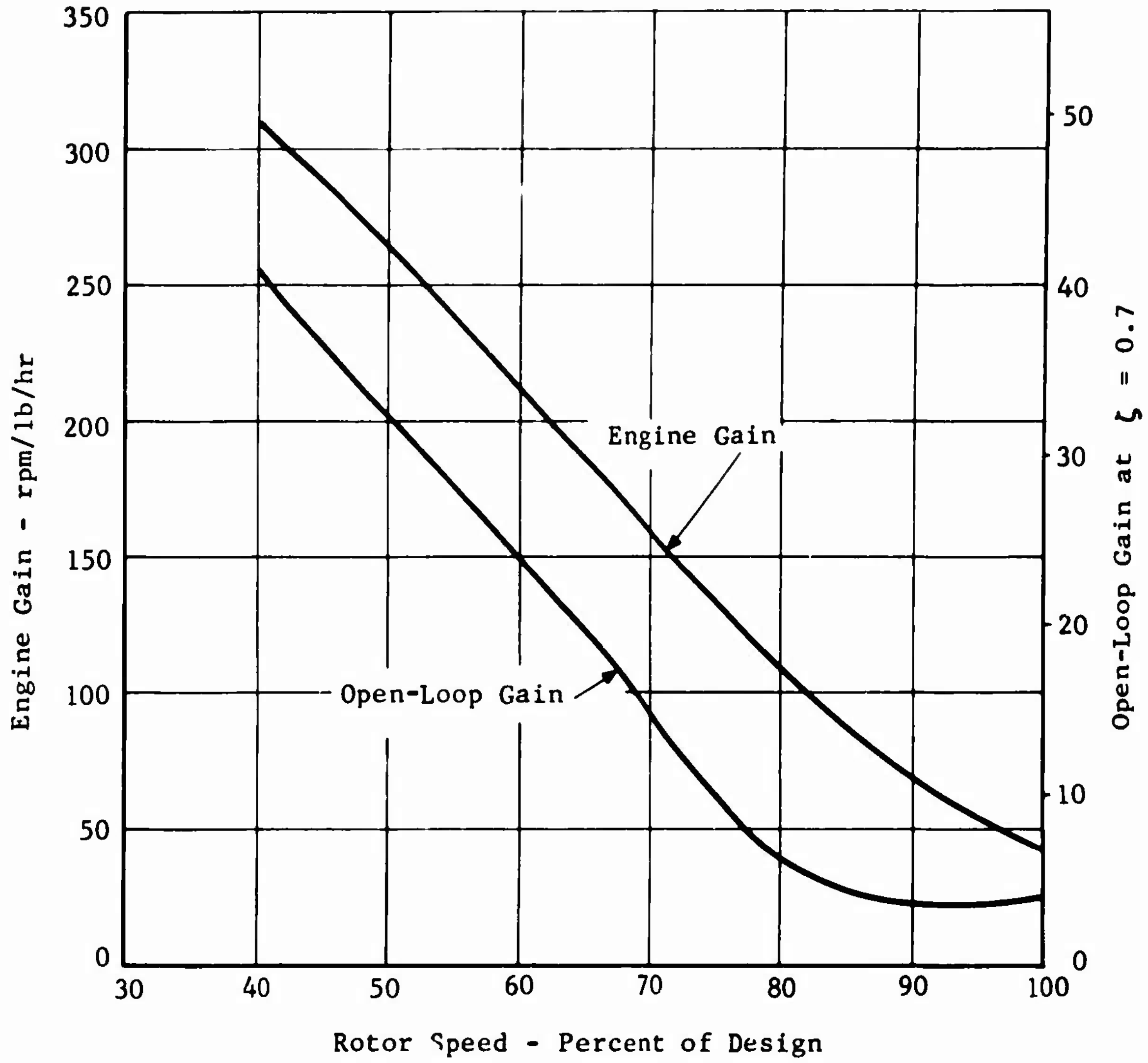


Figure 43. Engine Gain Compared to Open-Loop Gain of Speed Loop Required to Maintain Damping Ratio at 0.7.

Figure 44. Hybrid Computer Facilities.

The computer simulation diagram is shown in Figure 45. Dynamics of the individual components are given below:

Speed Sensor

Time Constant 0.01 sec
Time Delay 0.002 sec

Metering Valve

Time Constant 0.02 sec

Burner

Time Delay 0.02 sec

Temperature Sensor

Time Constant Classified
Time Delay Classified

Throttle

Time Constant 0.02 to 2 sec

The limiter shown on the fuel metering valve (Figure 45) ensures against flameout in the event of a rapid throttle reduction. This limiter sets the minimum rate of fuel flow through the valve when the throttle is being retarded between the idle and 100-percent range. The low limit chosen for this simulation was 20 pounds per hour.

Simulation Results

The results showed that safe accelerations can be obtained with a very simple system (see OTHER APPLICATIONS), and there is evidence that significant savings in cost and system simplification would be possible, depending on the mission. Drastic savings in propulsion system weight can also be realized because the fluidic fuel control system does not depend upon accessory drive train reduction gears for its operation.

The primary objective of the simulation was to synthesize the simplest control system capable of rapidly accelerating the engine from idle to 100-percent speed while still remaining under the surge line. First simulation results showed that the proposed control configuration became unstable when the temperature limiting control was added to the proportional-plus-integral speed control, although each loop by itself was stable. A method of stabilization was attempted and found to be successful. In the presence of large speed errors, the gain of the speed loop was decreased by a factor of 10, thereby decreasing the gain requirements of the temperature loop. This method of stabilization was chosen because it was in keeping with the simplified fuel control concept and would be easy to mechanize with fluid elements.

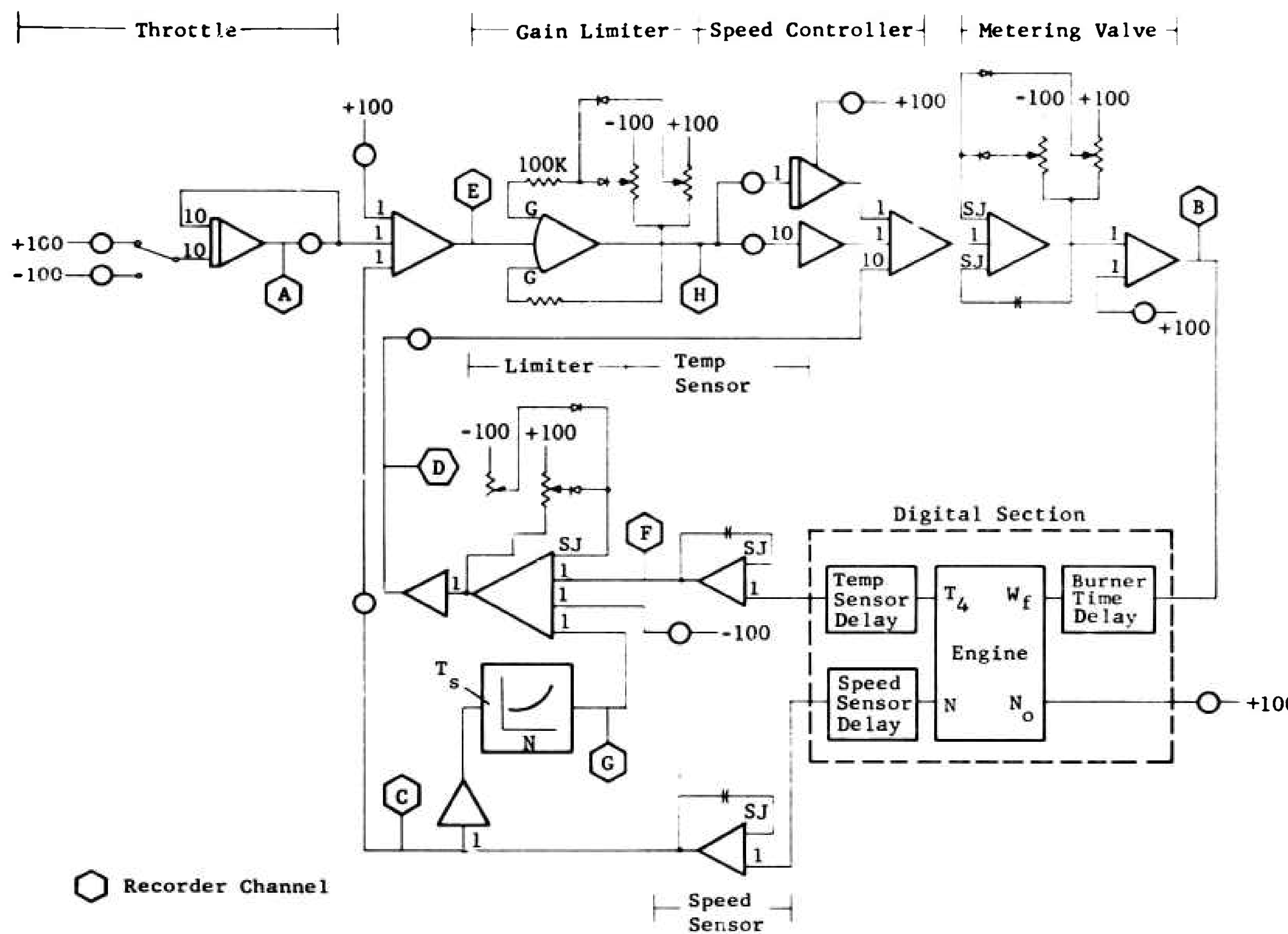


Figure 45. Small Gas Turbine Engine Computer Diagram.

The simulation objective was accomplished. Figure 46 is a photograph of an oscillographic display showing an acceleration from idle to 100-percent speed. The slight distortion of the axis and the scales results from the fact that the face of the oscilloscope tube is curved. A step command was put in through a 0.020-second throttle lag. The engine trajectory quickly drives above the set temperature line (not displayed on the oscilloscope), activating the temperature-limiting loop and reducing fuel flow after the manner of a limit control.

At the lower end of the speed range, fuel flow rate shows an underdamped response about the set temperature line. The response is damped out after engine speed reaches 73 percent.

Figure 47 shows the outputs at several points in the control system. Locations of the monitoring points are shown in the computer diagram, Figure 45. Fuel flow rate, turbine inlet temperature, and temperature error exhibit underdamped responses until 73-percent speed is reached. However, the simulated machine accelerates rapidly and smoothly with no evidence of oscillation in the speed curve.

It is probably impossible to arrive at an intuitive understanding of the dynamic response by looking at the steady-state curves which are put into the computer. However, a little appreciation for the reasons behind the response shown by the engine trajectory can be gained by reviewing the equation for GH and Figure 43. The equation for GH shows that the open-loop gain of the speed loop is directly proportional to engine gain K_N.

Figure 43 shows how the engine gain drops steadily as speed increases from idle to 100-percent speed. Above 73-percent speed, the combination of the fast engine response and rapidly rising set temperature line (see Figures 48 and 49) is believed to contribute to the damped response. The significance of Figure 49 is further explained under <u>Acceleration Times</u>.

Another way of looking at it is to note that the components making up the open-loop gain are all constant (when there is no error signal limiting) except for the engine gain K_N and engine time constant τ_e.

It has been previously noted that the engine gain changes by a factor of 7 whereas a factor of 10 change in open-loop gain is required to maintain a constant damping ratio. Thus, the fixed gain control does not maintain a constant damping ratio. In fact, the damping ratio of 0.7 occurs at only one speed (i.e., engine gain). At higher speeds the damping ratio is greater than 0.7 (system overdamped) and at lower speeds the damping ratio is less than 0.7, giving rise to an underdamped response.

The response to command for small speed changes was checked over the entire operating range, and the results are shown in Figures 50, 51, and 52. An underdamped response characteristic comes into evidence at the lower ranges of speed. If constant damping over the entire operating range were desired, a gain changer operating as a function of speed could be added to the loop, as is being done in more sophisticated systems.

The control system was also checked for deceleration capabilities and was found to work very well. A rapid deceleration from 100- to 80-percent speed is shown in Figure 53.

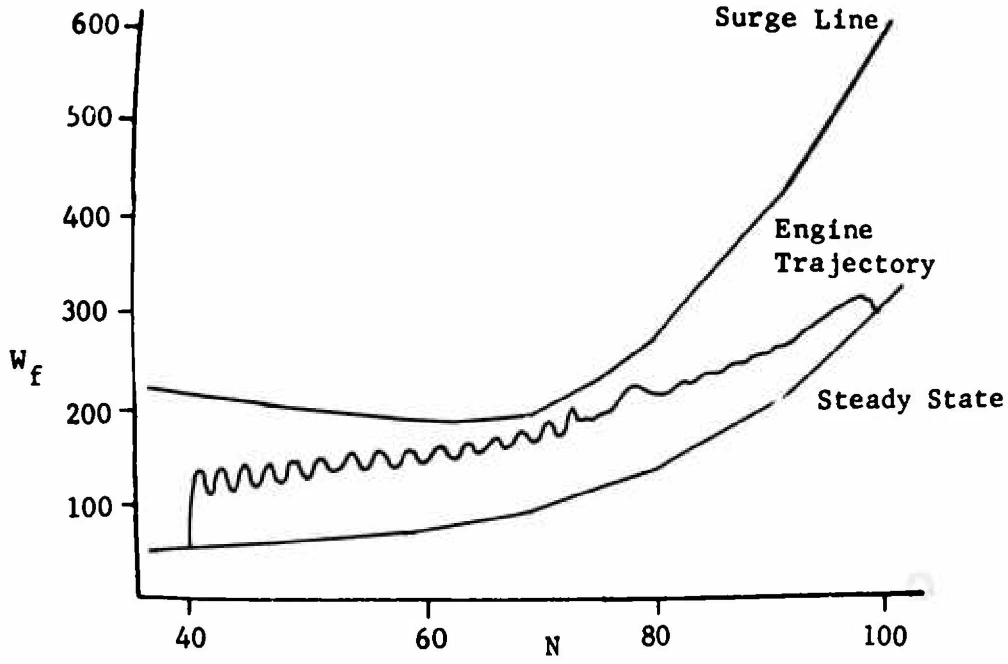


Figure 46. Engine Acceleration Trajectory for 100-Percent Speed Command From Idle.

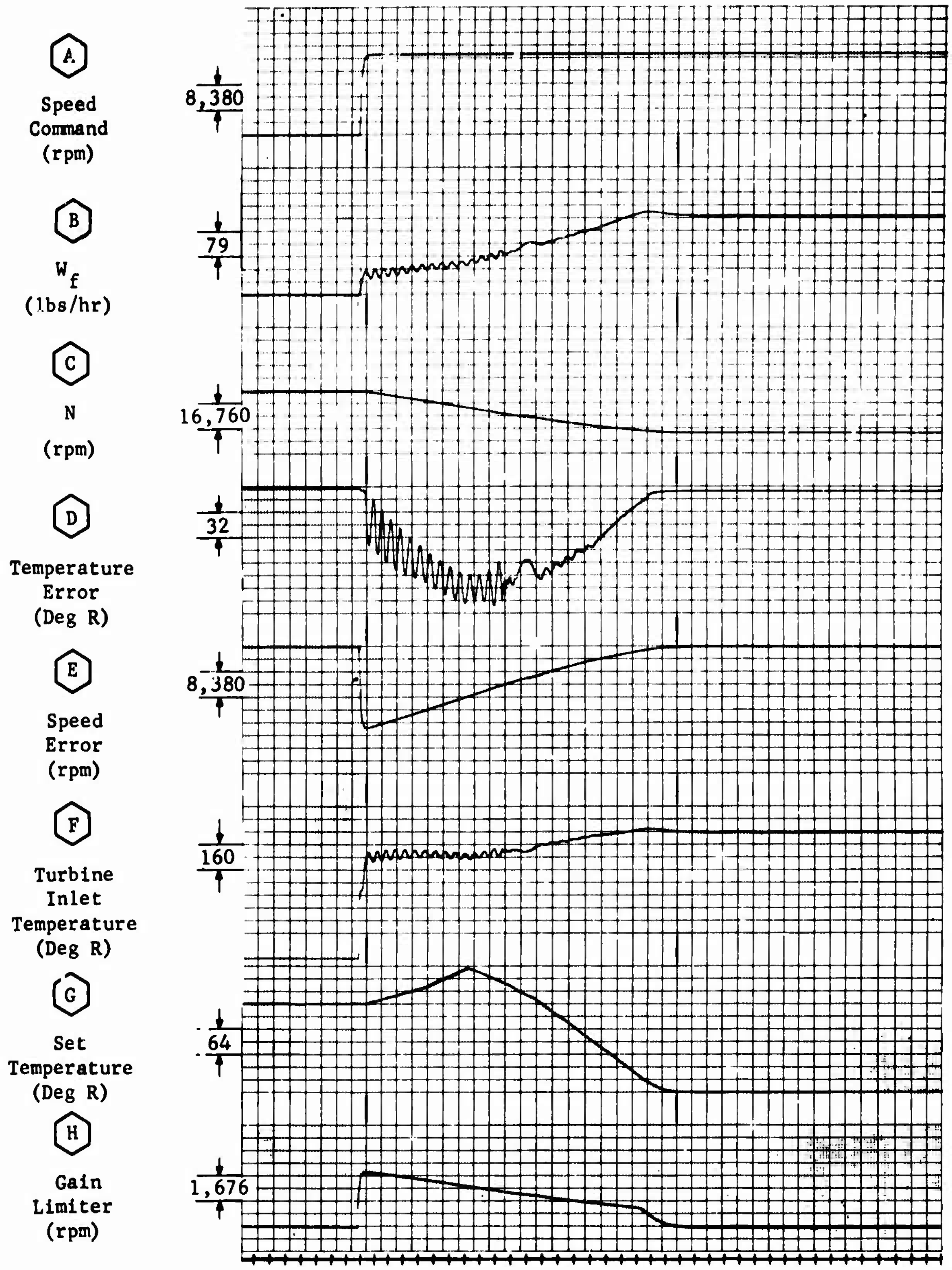


Figure 47. Computer Outputs for 100-Percent Speed Command From Idle.

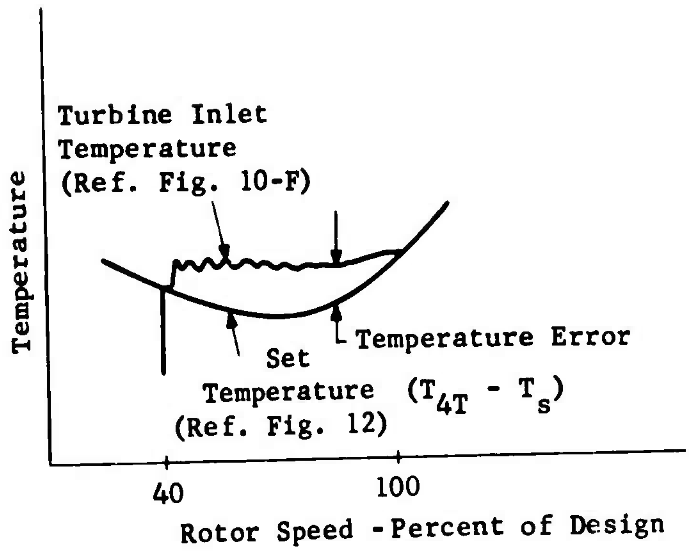


Figure 48. Typical Temperature Error Plot for 100-Percent Speed Command From Idle.

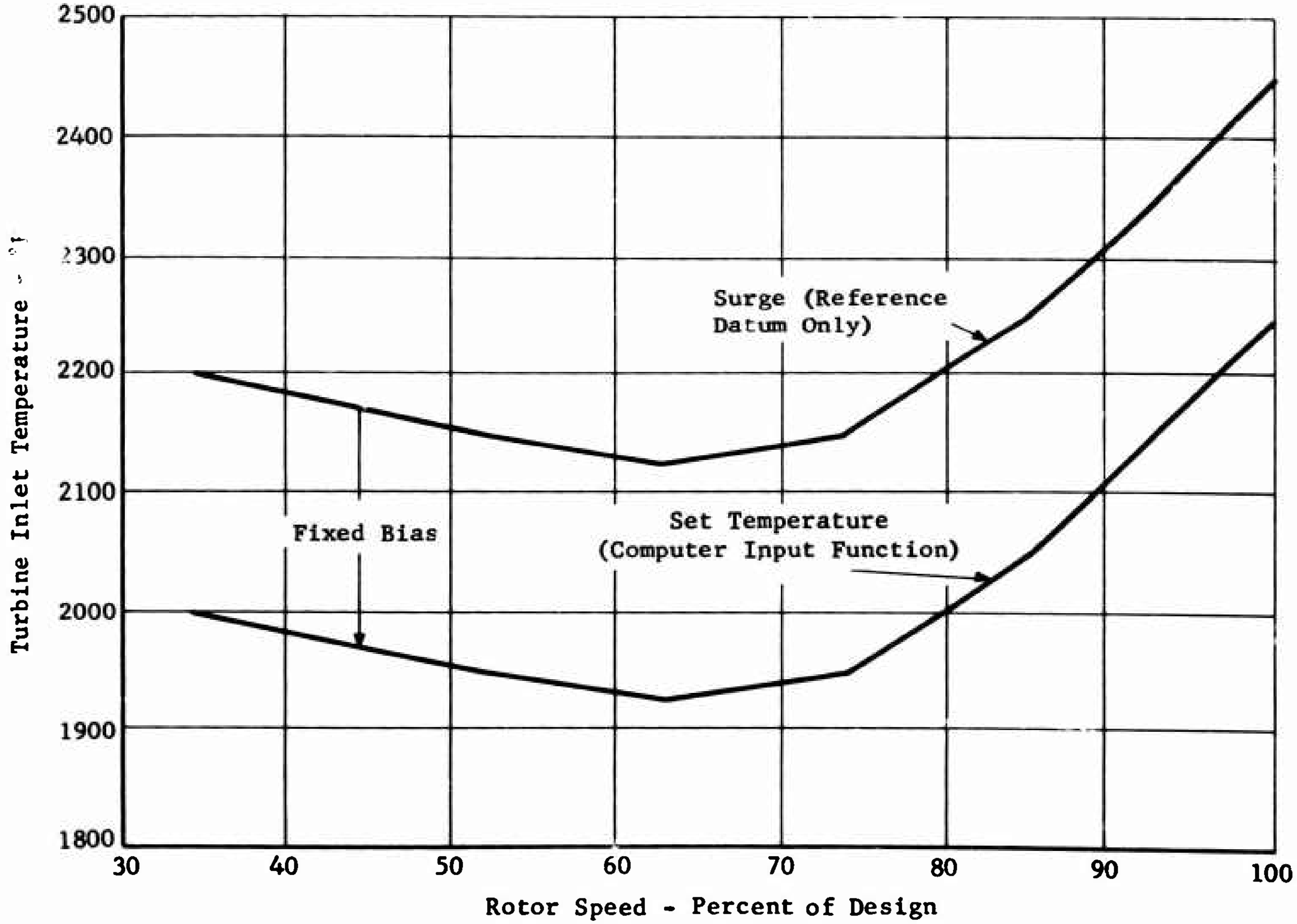


Figure 49. T Versus N Plot Showing Computer Function Generator Input to Temperature Controller.

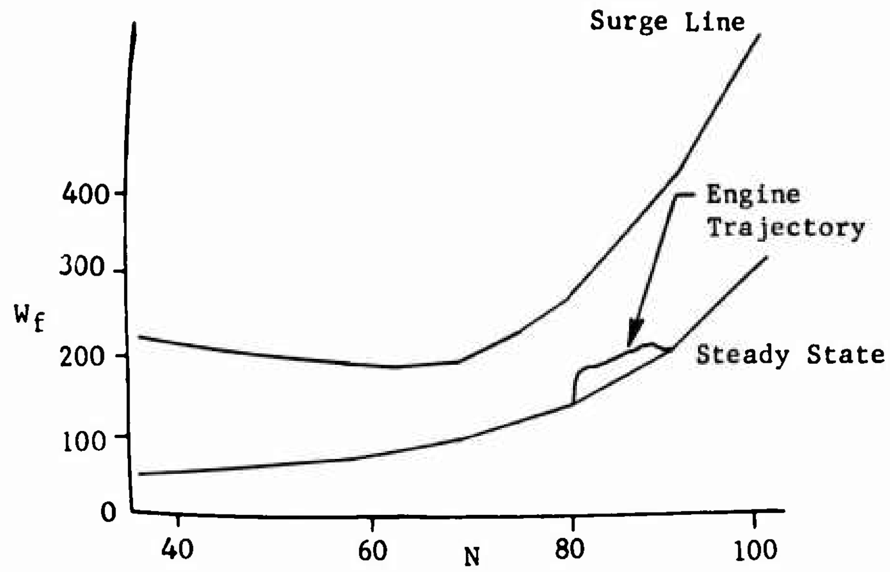


Figure 50. Engine Acceleration Trajectory for 90-Percent Speed Command From 80-Percent.

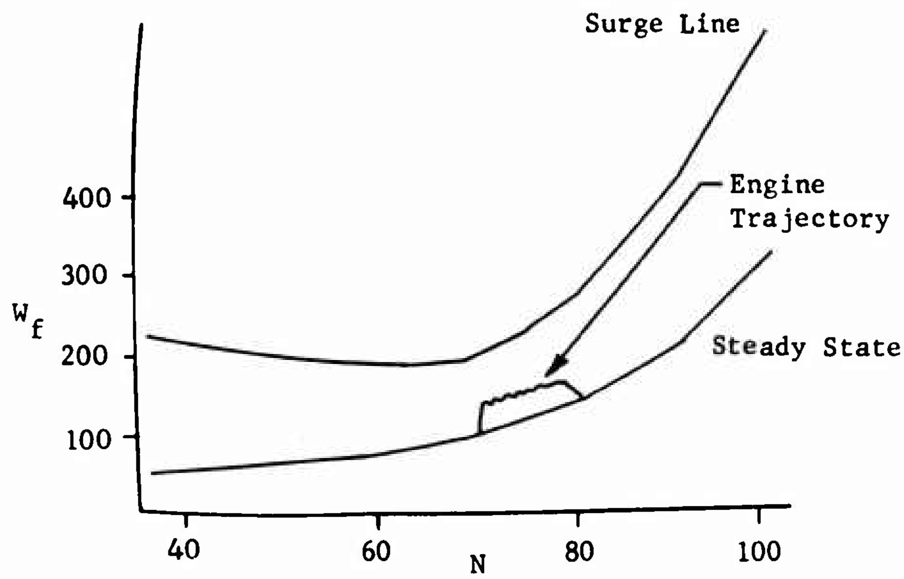


Figure 51. Engine Acceleration Trajectory for 80-Percent Speed Command From 70-Percent.

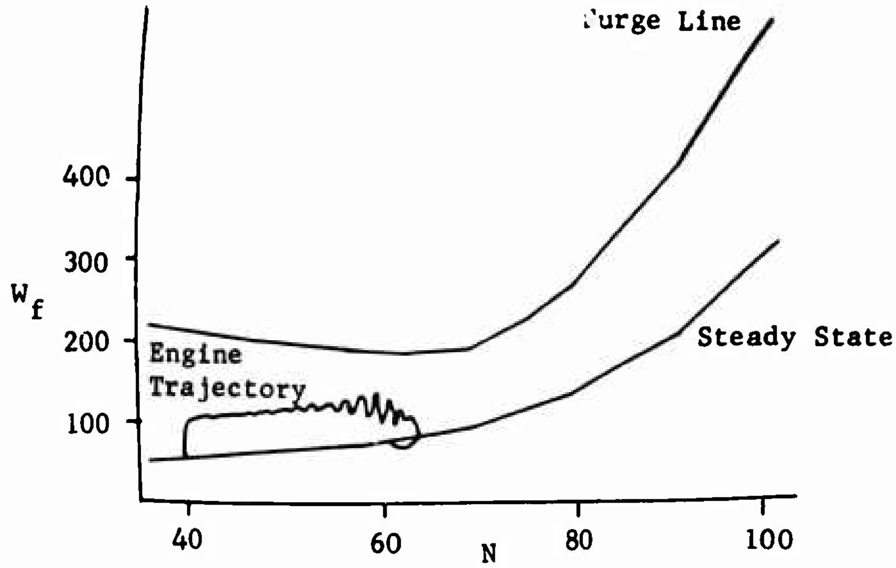


Figure 52. Engine Acceleration Trajectory for 60-Percent Speed Command From Idle.

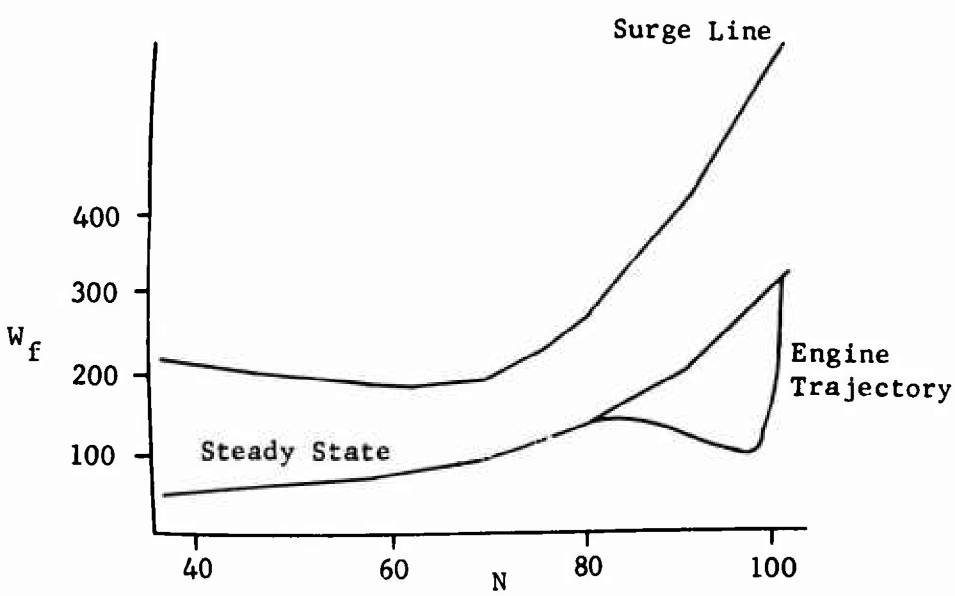


Figure 53. Engine Deceleration Trajectory From 100-Percent to 80-Percent.

Acceleration Times

The time required to accelerate the engine to a new commanded speed was obtained by recording engine speed from the computer. Accuracy of these time values is as good as the accuracy of the engine map data (see Simulation Approach). Compared to time values attained with a map constructed from actual test data on a small engine, these computed time values probably differ by less than +30 percent. Table IV lists the computed acceleration time required to cover 95 percent of a commanded speed change.

TABLE IV
ACCELERATION TIMES FOR SPEED COMMAND CONTROL WITH
0.02-SEC THROTTLE LAG AND TEMPERATURE LOOP

Acceleration Speed Range-0.02-Sec Lagged Step Command (% of Design Speed)	Acceleration Time ±30%* (sec)
Full Range	
40 - 100	3.8
20-Percent Range	
40 - 60	1.9
80 - 100	2.8
10-Percent Range	
70 - 80	1.6
80 - 90	1.9
90 - 100	2.3
Deceleration	
100 - 80	2.0

*Time required to cover 95 percent of tabulated commanded speed change.

The simulated acceleration times can be substantially improved, particularly at the higher engine speeds, by a simple refinement of the simulation. This can be better understood by explaining how the set temperature function was generated for the present simulation.

To save time and simulation cost to the program, the set temperature line was generated simply by biasing the surge-line output of the computer function generator as shown in Figure 49. The bias was adjusted until the displayed engine trajectory line (see Figure 46) stayed below the surge line during acceleration.

Figure 54 shows the relationship between the set temperature line utilized in the control system and the surge line used as a reference datum and for visual evaluation of the control simulation. Due to the translation from T-N coordinates in Figure 49 to W_f-N coordinates in Figure 54, the set temperature line in Figure 54 does not closely track the surge line. Although this was the only set temperature line tried in the simulation, it appears that further improvement in acceleration time could be achieved by optimizing the shape of the set temperature line at no additional cost in added system complexity.*

OTHER APPLICATIONS

The control system configuration will largely be dictated by the engine mission. There may be instances where the small gas turbine engine is used essentially as a constant speed machine. An example of this would be an APU for standby power generation. For an application of this type the control system need only have a good speed controller. The surge and overtemperature protection could consist simply of a lag built into the throttle so as to limit commanded acceleration to a value below the surge and overtemperature limits. This type of control system was checked on the hybrid computer and was found to work very well with a throttle lag of 2 seconds. The control system was the same as the proposed configuration except that the temperature loop was omitted.

The oscilloscope display for acceleration from idle to 100-percent speed is shown in Figure 55. This system also works well for smaller commands as shown in Figure 56. Although the time to accelerate from idle to 100-percent speed is about two and one-half times longer than with the proposed configuration, as shown in Table V, this type of system would be very satisfactory where there is no need for rapid acceleration. In some cases the control simply needs the capability of controlling the engine to constant speed and protecting the machine from heavy-handed throttle operation.

*The A.F. integrated engine control contract has in the temperature loop a proportional-plus-integral controller plus temperature rate limiting, enabling it to ride closely the surge line during acceleration. While a faster acceleration is provided, the system is also more complex. Pressure, altitude and Mach number compensation are also being designed into the U.S.A.F. system for wind tunnel testing at altitude.

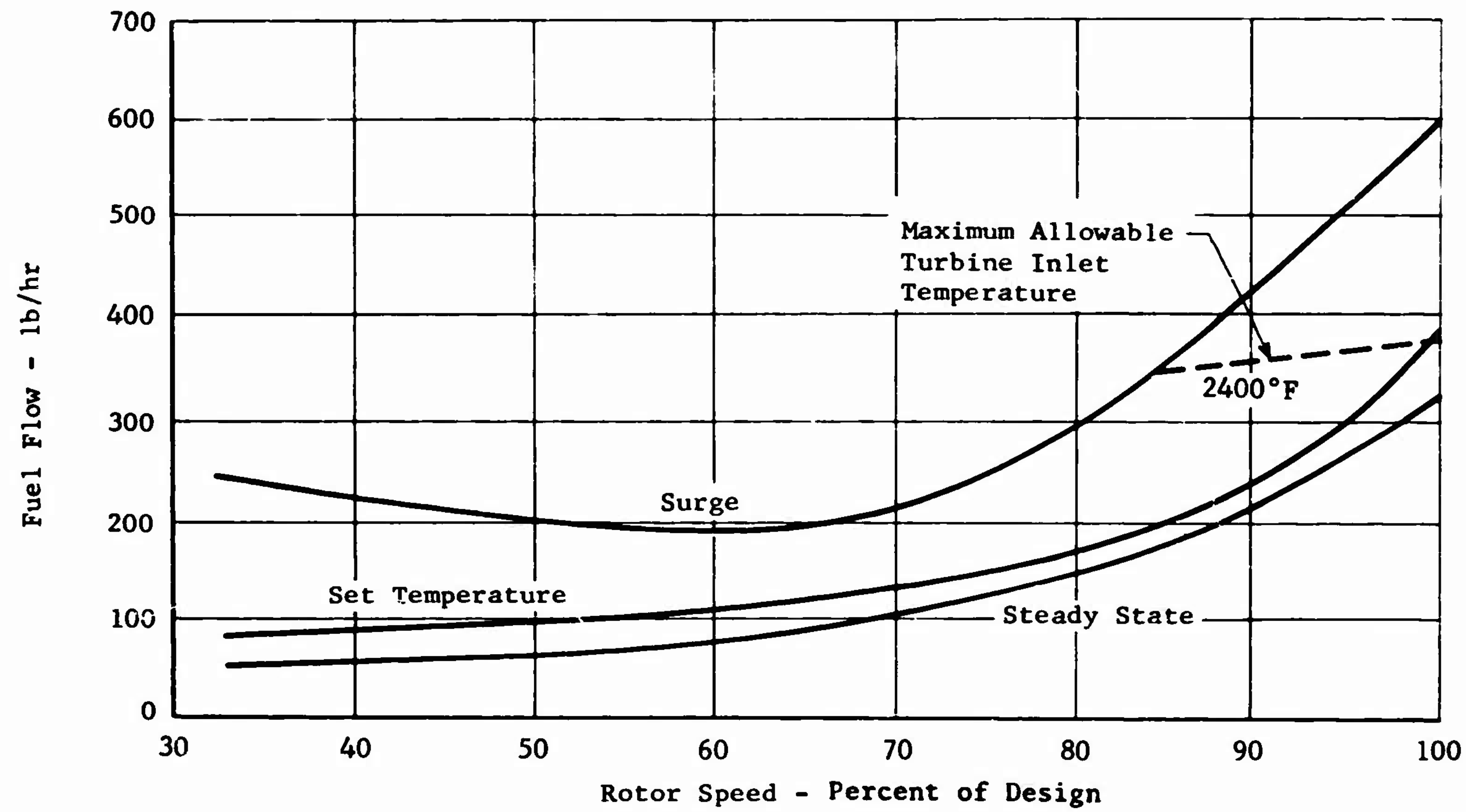


Figure 54. W_f Versus N Plot Showing Relationship of Set Temperature to Surge Line.

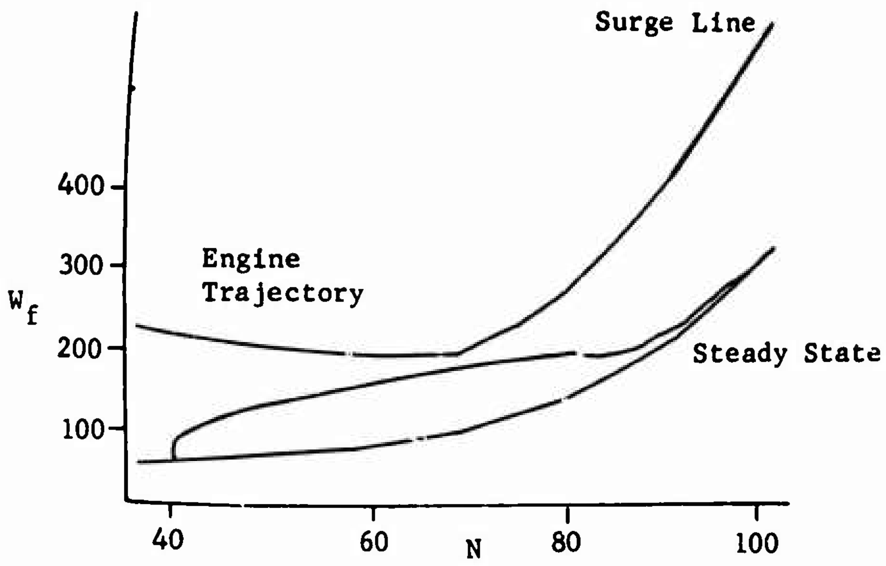


Figure 55. Acceleration to 100-Percent Speed From Idle - With 2-Second Throttle Lag and No Temperature Loop.

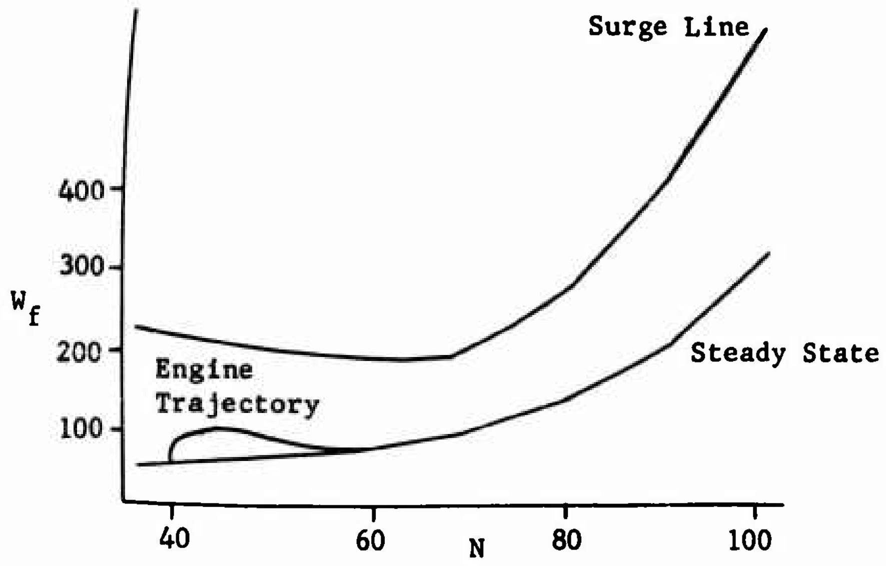


Figure 56. Acceleration to 60-Percent Speed From Idle - With 2 Second Throttle Lag and No Temperature Loop.

TABLE V
ACCELERATION TIMES FOR SPEED COMMAND CONTROL WITH
2-SEC THROTTLE LAG AND NO TEMPERATURE LOOP

Speed Range - 2-Sec Lagged Step Command (% of Design Speed)	Acceleration Time ±30%* (sec)
Full Range	
40 - 100	10.0
20-Percent Range	
40 - 60	6.8

*Time required to cover 95 percent of tabulated commanded speed change.

Some applications such as VTOL may require fast response only in the top 10 or 15 percent of the engine speed range. In this region it is very difficult to encounter surge without first overheating the engine (see Figure 54). For this application, therefore, a simpler control is possible in that the set temperature generator and corrected speed computer (see Figure 40) could be replaced by a fixed temperature limit and two idle speeds - one a low level idle and the second a high level idle at about 85-percent engine speed. Acceleration from one idle to the other could be accomplished by a self-contained engine sequence control included as part of the starting system.

Fluid control systems are inherently simpler than hydromechanical controls of similar sophistication. For any given control system approach, however, the complexity increases when the engine mission requires it to accelerate from idle to 100-percent speed in the shortest possible time. This application places a premium on the dynamic response characteristics of the control system.

The ultimate in response will be achieved by the control system which is able to ride closely the surge line and maximum turbine inlet temperature line with little or no overshoot. The Honeywell fluidic turbine inlet temperature sensor has a time constant less than one-tenth that of a compensated thermocouple used at the turbine inlet. It is believed that employment of this sensor to determine the surge line and temperature limit line in a sophisticated system will provide an acceleration response limited only by the machine itself.

In summary, Table VI lists four systems of the Speed Command Type in order of increasingly rapid response. A rough comparison of complexity is given by listing the number of functional blocks in the system block diagram. For this study, only Systems 1 and 2 were simulated. Systems 2 and 3 are essentially the same, the only difference being the method of shaping the set temperature line. During the first quarter of 1966, the simulation of System 4 will be completed under the Integrated Engine Control Contract.

TABLE VI
GENERAL COMPARISON OF SPEED COMMAND CONTROL SYSTEMS

Item No.	Relative Response (compared to No. 1)	System Description	Relative Complexity* (estimated number of functional blocks)
1 - Speed Control with Long Throttle Lag	Slow - (Accel. time = t)	Proportional-plus-integral speed control, 2-sec throttle lag.	11
2 - Speed Control with Simple Temp. Limiting	1/3 to 1/4 (t)	Proportional-plus-integral speed control loop, temperature limiting loop, 0.02-sec throttle lag; set temperature line shaped to surge line on T-N coordinates.	19
3 - (Same as No. 2)	About 1/4 to 1/5 (t)	Same as No. 2, except set temperature line shaped to surge line on W_f-N coordinates.	19
4 - Speed Control with More Complex Temp. Loop	Limited only by engine	Proportional-plus-integral speed loop, proportional-plus-integral plus rate temperature loop, 0.02-sec throttle lag; controls very close to surge line.	25

*These estimates apply to sea level static systems with no provision for variable compressor geometry or variable exhaust nozzle area. The temperature control loops were each assumed to include four temperature sensors.

START AND RELIGHT SEQUENCE CONTROL

TECHNICAL APPROACH

The approach most suitable for automating the starting and airborne relight sequence depends on the engine application. Of principal importance is the question, "How long can we afford to wait before attempting a relight in event of flameout?" Once this question is answered, the relative complexity of the system will be determined by the degree of reliance placed on the skill and alertness of the operator and the relative importance of weight and cost economics.

To facilitate study of the requirements for a fluidic engine sequence control, a Light Observation Helicopter (LOH) application is assumed as a typical application for the small engine. The operator skill level is assumed to be commensurate with present pilot selection and training practices. Where a reasonable time period is available for response, the monitoring capability of the pilot is utilized to permit simplification of the control system. Thus, during startup on the ground, it is suggested that the pilot be relied upon to monitor oil and fuel pressure rather than incorporating automatic lockout circuits for engine protection.

The proposed approach to the engine starting and relight system permits the pilot to start the engine simply by setting the throttle to IDLE position, then momentarily pushing the START button. The sequence control regulates the initiation and scheduling of fuel flow as a function of engine speed. Ignition is energized when a preselected speed is reached and is deenergized by a flame monitor.

The system also includes airborne relight capability. The sequence control automatically energizes ignition upon flameout, if the engine is at running speed and the throttle is not in the OFF position. A time delay circuit cuts off the ignition after a preset time if the engine fails to relight. Further relight attempts must be made manually using air-start procedures.

Both the start and relight modes minimize the duration of ignition operation. This approach will extend the life of the ignition system while retaining the instantaneous relight characteristic of a continuous ignition system.

The block diagram of a possible engine starting sequence control is shown in Figure 57 . Blocks in solid outline depict component functions required specially for the engine starting and relight control. Blocks in dashed outline are components already required for the fuel control system. The following description of the starting and relight sequences applies to both the block diagram of Figure 57 and the schematic diagram of Figure 58.

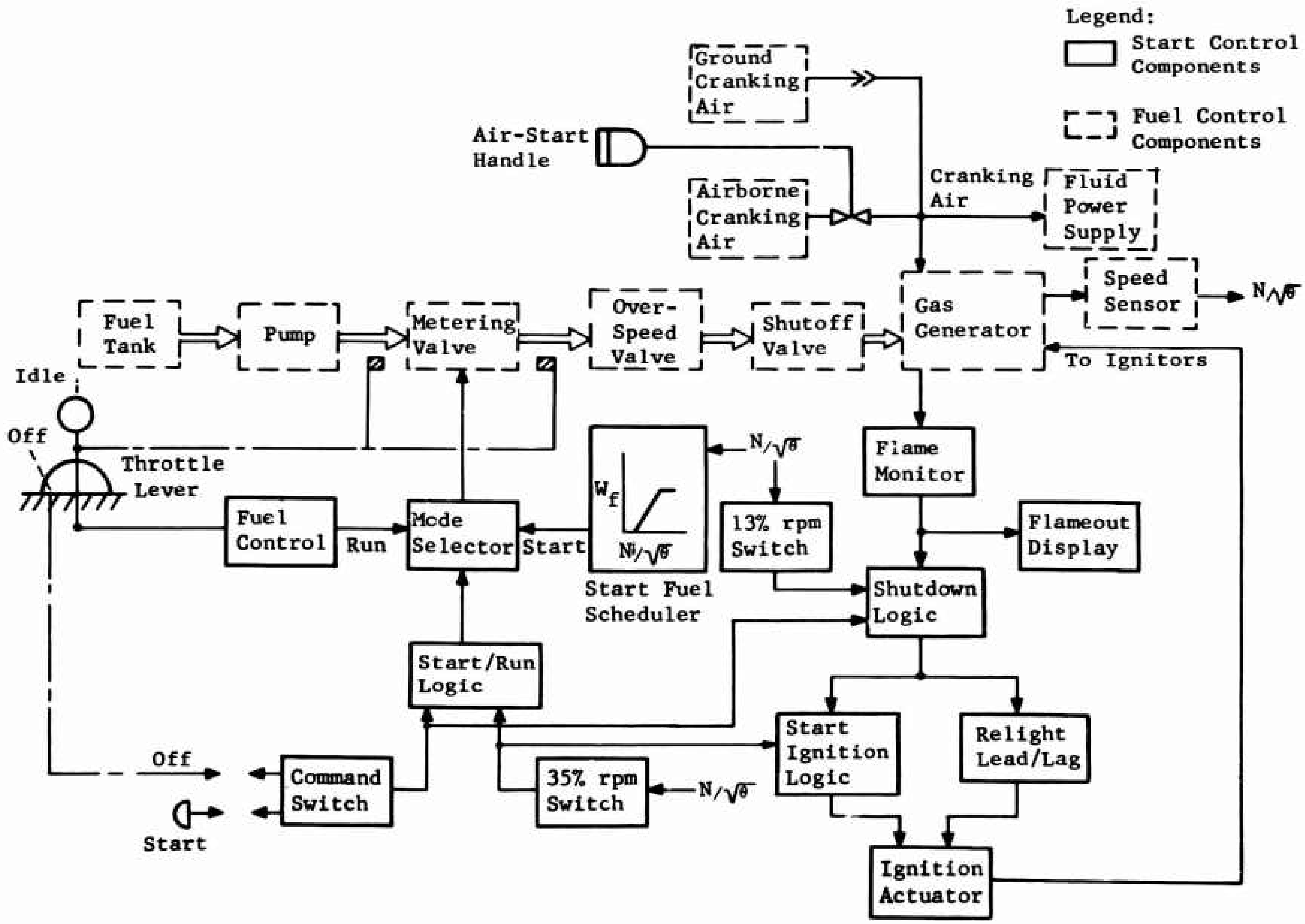
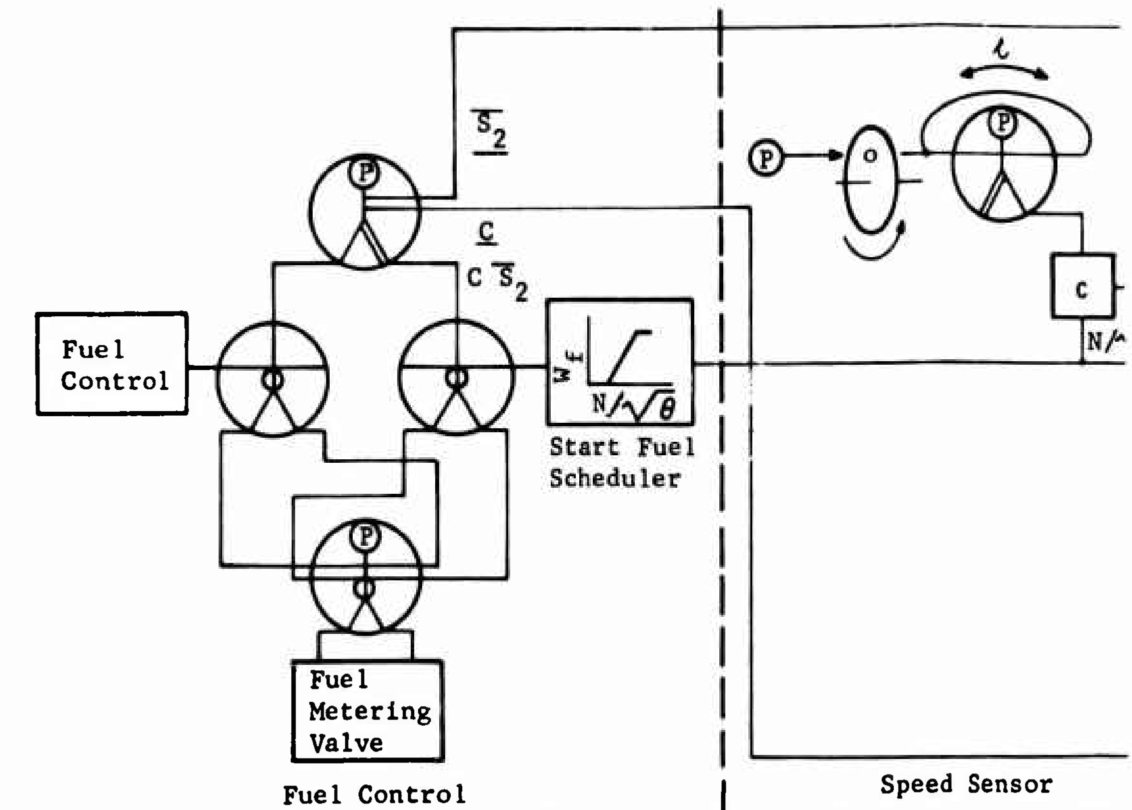


Figure 57. Start and Relight Sequence Control.



Mode - Start & Relight Controls
Both Shown In Ignitors - Energized Mode

Pressure State

( ) = Pressure Signal   ( ) = No-Pressure Signal

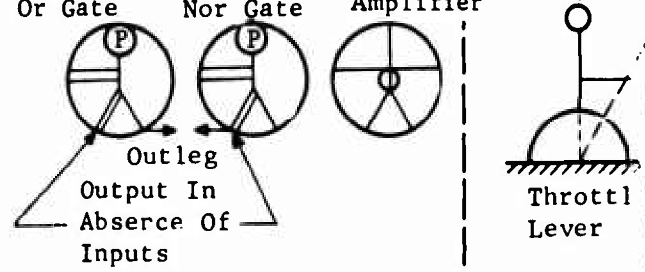
Logic Function

C = Start Command
F = Flame
I = Ignitors Energized
L = Lead/Lag Flameout Function
S_1 = Speed ≥ 13% rpm
S_2 = Speed ≥ 35% rpm

Complementary Function

$\bar{C}$ = Stop Command
$\bar{F}$ = Flameout
$\bar{I}$ = Ignitors Deenergized
$\bar{S}_1$ = Speed < 13% rpm
$\bar{S}_2$ = Speed < 35% rpm

Legend

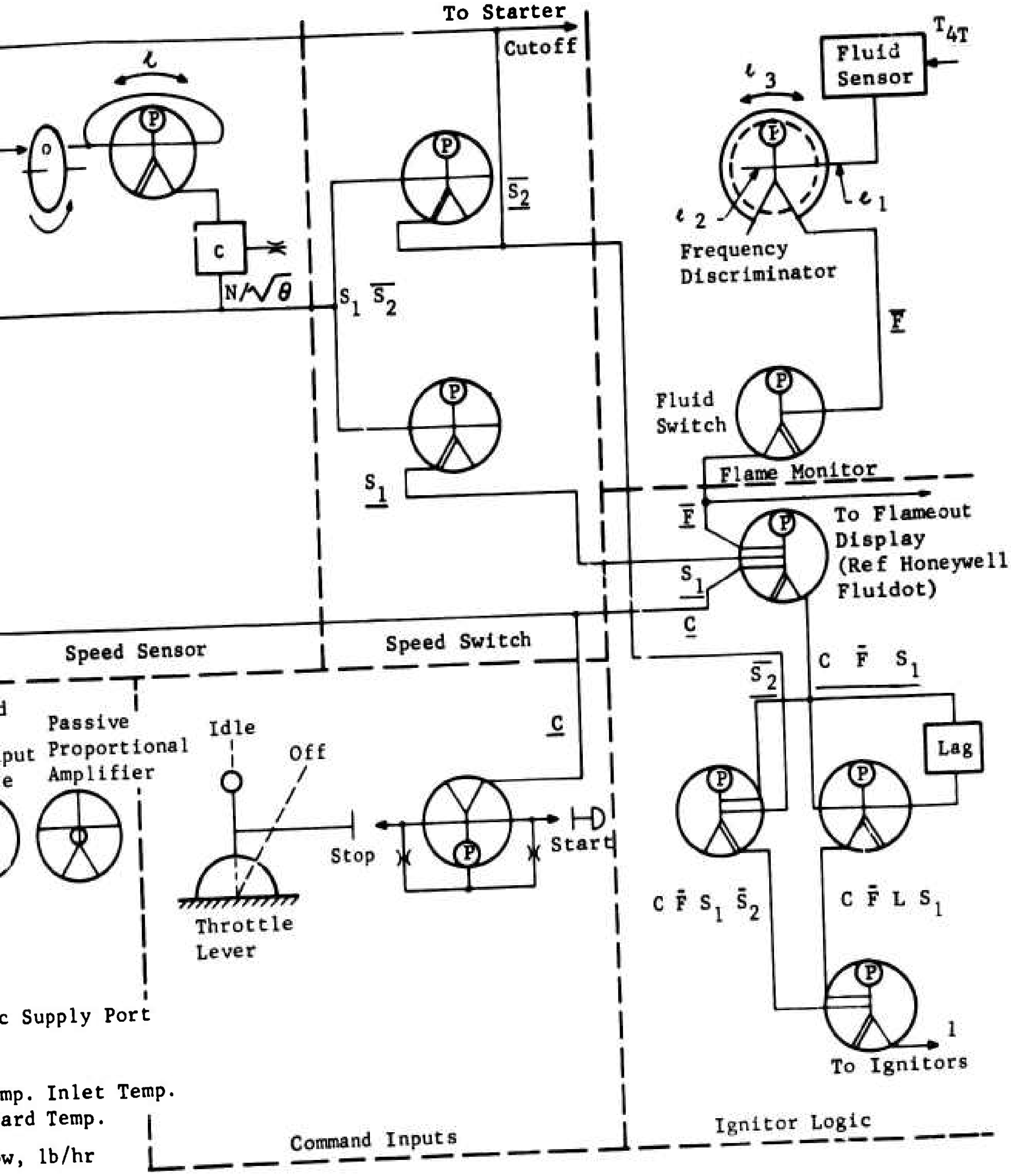


Ⓟ = Pneumatic Supply Port
N = rpm
θ = Ratio Comp. Inlet Temp. to Standard Temp.
W_f = Fuel Flow, lb/hr
T_{4T} = Turbine Inlet Temp.
ℓ = Length

Logic Equations

$I = C\bar{F}S_1\bar{S}_2 + C\bar{F}LS_1 = C\bar{F}S_1(\bar{S}_2 + L)$ $\bar{I} = \bar{C} + F + \bar{S}_1 + S_2\bar{L}$

Figure 58. Start and Relight Sequence Control.



mp. Inlet Temp.
ard Temp.
w, lb/hr
Inlet Temp.

$' + \bar{S}_1 + S_2\ \bar{L}$

B

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GROUND STARTING SEQUENCE

With the throttle in the OFF position, the MANUAL SHUTOFF valve is opened and compressed air is applied for cranking the engine. The compressed air supply also furnishes air to the FLUID POWER SUPPLY for powering the fluidic control elements.*

As the rotor begins to spin up, the pilot advances the throttle directly to the IDLE position and momentarily presses the START button. The START button energizes the COMMAND SWITCH which provides a command logic input to the START/RUN LOGIC and the SHUTDOWN LOGIC. The START/RUN LOGIC operates the MODE SELECTOR which selects the source of command for the METERING VALVE. The START/RUN LOGIC circuit automatically holds in the START MODE until the corrected-speed signal reaches idle, at which time the 35 percent rpm SWITCH causes the logic to switch to the RUN mode.

A second function of the COMMAND SWITCH is to arm the SHUTDOWN LOGIC circuit. When a preset speed of 13 percent rpm is reached, the command signal input to the SHUTDOWN LOGIC permits the FLAMEOUT MONITOR to energize the START IGNITION LOGIC and hence the IGNITION ACTUATOR. The START IGNITION LOGIC passes the ignition signal only if there is a START command, <u>and</u> there is no flame, <u>and</u> rotor speed is greater than 13-percent rpm, <u>and</u> rotor speed is less than 35-percent rpm. Unlike the RELIGHT LEAD/LAG, the START IGNITION LOGIC holds the ignition on until a flame is sensed.

CONTROL OF STARTING FUEL

Reviewing the starting fuel control functions in greater detail, the START/RUN LOGIC circuit switches the MODE SELECTOR to the START mode whenever a start is commanded and engine speed is less than idle rpm. Fuel rate is then commanded by the open loop START FUEL SCHEDULER rather than by the normal FUEL CONTROL. This is necessary because the speed command signal from the throttle is calling for idle rpm, but the slowly accelerating rotor cannot immediately respond. The resulting closed loop error signal from the FUEL CONTROL would tend to drive the METERING VALVE hard open. However, by switching in the open loop START FUEL SCHEDULER, a scheduled fuel flow is provided for a given corrected speed signal. When idle speed is reached, the MODE SELECTOR is switched back to the RUN mode by the

*Starter cutoff for the J65 engine occurs at a speed approximately one-half of idle rpm and a CDP of 2 to 2.5 psig. It takes 16 to 20 seconds after starter cutoff before this engine reaches idle speed, and a CDP of 6 psig, which is necessary to power the fluidic elements.
<u>Conclusion</u>: In order for a fluidic start sequencing system to function satisfactorily, either the starting compressed air supply must be left on for 20 seconds after starter cutoff, or fluidic elements will need to be designed to function satisfactorily on the supply pressure existing at the time of starter cutoff. Until a sequence control is worked out for a specific application it will be assumed that compressed air will be supplied to the fluid control elements until the engine has accelerated to idle speed.

START/RUN LOGIC. Fuel flow rate from that point on is controlled by the normal FUEL CONTROL.

Although the START FUEL SCHEDULER operates open loop, it is believed that no adjustment will be necessary to accommodate various fuels, on the basis that only the starting time will be affected. This point should be confirmed by further investigation.

ENGINE SHUTDOWN

If the pilot observes unsatisfactory oil or fuel pressure or simply wishes to shut down the engine, he retards the throttle to the OFF position, an action which puts a STOP command into the COMMAND SWITCH. This serves both to shut off fuel flow and to lock out the ignition circuit, as follows: the OFF command locks the MODE SELECTOR in the RUN mode. This locks out the START FUEL SCHEDULER, permitting the FUEL CONTROL to drive the METERING VALVE fully closed. The OFF command input to the SHUTDOWN LOGIC locks out the ignition circuit so that a subsequent flameout signal cannot actuate the ignitors as the rotor runs down.

AUTOMATIC RELIGHT SEQUENCE

An automatic relight feature is particularly useful in those applications where

1. Constant ignition is not employed.
2. The application does not permit waiting for flameout to manifest itself in loss of power before initiating a time consuming manual restart.
3. The type of vehicle employed and/or the use of a free coupled turbine drive makes it impossible to start by windmilling.

One possible automatic relight system, shown in Figures 57 and 58, utilizes a sensor which could be based on the flame sensitive powertube or turbine inlet temperature sensor. When flameout occurs, a signal from the FLAME MONITOR is fed to the FLAMEOUT DISPLAY and the SHUTDOWN LOGIC. If the engine speed is above 13-percent rpm (and the START command prevails), the flameout signal is allowed to pass through the SHUTDOWN LOGIC to the RELIGHT LEAD/LAG. The RELIGHT LEAD/LAG first provides an output signal to the IGNITOR ACTUATOR. If the engine fails to light after a predetermined time period, the RELIGHT LEAD/LAG cuts off the relight signal.

At the altitude at which flameout occurs, the ignitors are automatically energized for a brief period of time. During this time period, the rate of fuel flow through the metering valve will be about the same as the fuel rate existing at the instant of flameout. Thus, it is possible that no special provision need be made for controlling fuel flow during relight.

If relight does not occur almost immediately, the ignitors are deenergized. The pilot may attempt further restarts manually if the aircraft carries airborne cranking capability as illustrated in Figure 57.

FLUID POWER SUPPLY FOR RELIGHT

In most installations it is assumed that compressor bleed air will be used to power the fluidic control elements. Engine speed must be approximately at idle or above in order to provide the 5-psig CDP required by the fluidic elements. In a multiengine installation, the fluid power supply would be assured of adequate air pressure in spite of failure of one engine.

Even in a single-engine installation, it appears from Figure 59* that adequate CDP is available for longer than is necessary for automatic relight purposes. Thus, choice of fluidic versus electronic or pneumatic** methods of flameout sensing can be based on factors other than the loss of power to the fluidic elements.

SELECTION OF FLUID LOGIC ELEMENTS

A recommended approach to fluidic mechanization of the start and relight control is given in the schematic diagram shown in Figure 58. Honeywell development work has concentrated on the wall attachment amplifier, and this is the type of element employed throughout the suggested sequence control. The logic is based upon the NOR element, since AND elements of the wall attachment type have not yet been developed to the point of reliable operation in the face of varying pressure conditions.

As shown in the legend of Figure 58, the NOR element is biased so that, in the absence of control pressure inputs, the output will switch to a preferred outleg. In a dual input NOR element, it is necessary that neither the one control input nor the other be present if the element is to be permitted to switch to the preferred outleg.

The same device becomes an OR element if the output signal is taken from the not-preferred outleg (opposite the control ports). In this case, the presence of either one control signal or the other will result in an output in the indicated outleg.

The "biased bistable" amplifier terminology indicated below is really monostable in the absence of control inputs; but since the biasing effect can be imposed by the associated circuitry outside the amplifier itself, the "biased bistable" term is used. In the absence of bias, the stage will be truly bistable, behaving as a flip-flop in that the output will remain in the outleg where last switched until an opposite control input is applied.

* Honeywell test data taken on a J85-7 engine, January 12, 1966.

**"Fluid Amplifier Control System for Advanced Turbojet Engines," Honeywell Proposal document R-ED 17002-1, 4 December 1962, to ASD. Reference CDP Flameout Control, page 42.

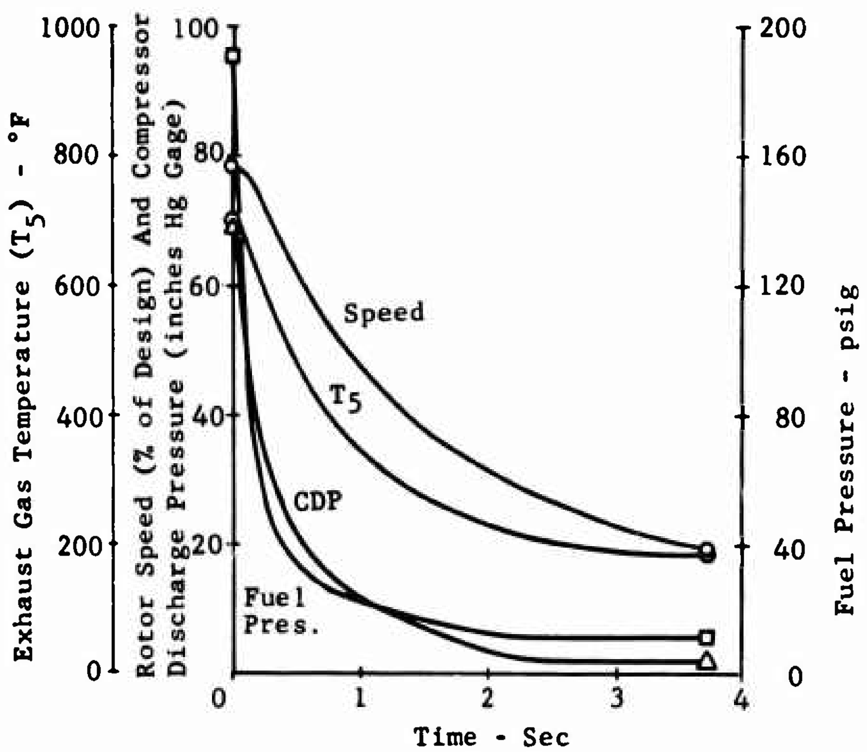


Figure 59. Variation of Engine Parameters After Fuel Cutoff at 78.3-Percent RPM (J85-7).

It is very likely that additional amplifiers beyond those shown in Figure 59 will be required to achieve the necessary gain and impedance matching. Considering only the basic functional stages shown in the schematic diagram, the quantity breakdown of logic and other fluidic elements is as follows:

Logic Elements	Quantity
Dual Input NOR	2
Triple Input NOR	1
Dual Input OR	1
Sensors, Summers, Switches, Timers, Etc.	
Bistable Amplifier	2
Biased Bistable Amplifier	5
Summing Amplifier	1
Passive Proportional Amplifier	2
Total	14

Logic equations for the start and relight sequence functions are given in the legend of Figure 58. An inferior-bar coding is used with the logic function symbols to designate pressure states throughout the circuit. Both start and relight controls are shown with instantaneous pressures appropriate to the "ignitors-energized" mode. Description of the operational functions is the same as that given in conjunction with Figure 57.

MANUAL OVERRIDE PROVISIONS

Safety practice requires that control of fuel flow rate be maintained in spite of failure of any single component in the fuel control system. The control system should be so designed that the pilot can command fuel flow if the fuel control fails in the closed position or can restrict fuel flow if the fuel control fails in the open position.

One method of achieving the desired end is to add a mechanical override linkage, illustrated schematically in Figure 57. The linkage drives a yoke assembly incorporated inside the pressure envelope of the fuel metering valve. In the emergency mode, the yoke clamps on the ends of the valve spool and the spool is mechanically positioned by the throttle-actuated linkage. In the normal mode the yoke is unclamped and located outside the displacement range of the spool. Frequency response is thus unimpaired.

By far the greatest single cause of catastrophic failures in servo valves* has been due to contamination in the hydraulic amplifier section. The ability to manipulate directly the servo spool is therefore of greater value than driving the nozzle flapper.

One possible way of engaging the override linkage is to incorporate an (H) slot in the throttle quadrant. The throttle lever is shifted from normal to override modes by moving the lever laterally from one leg to the other of the (H) slot. The principal advantages of this approach are that throttle control in the override mode is achieved with the same lever used in the normal mode and the linkage reduction can be designed so as to utilize approximately the full quadrant displacement angle.

*Electric torque motor failures are also significant but are not applicable to pneumatic input servo valve suggested for the engine control.

SUBSYSTEM AND COMPONENT DESCRIPTION

This section presents a brief review of unclassified features of fluidic components utilized in the simulated fuel control system. For additional details the reader is referred to classified do uments produced under Air Force Contract AF 33(615)-2696, such as Advanced Pneumatic Controls For Integrated Propulsion Systems, Semiannual Report 20268-SA1, 15 November 1965.

TEMPERATURE SENSOR

Sensor

The fluid temperature sensor is simply a temperature sensitive fluid oscillator. The hot gas whose temperature is to be measured impinges on a fixed splitter and is forced to oscillate between two cavities which provide a low impedance path for the hot gas. The cavity size sets the length of the path the gas will follow. Since the gas pulse will travel at the acoustic velocity, c, the oscillating frequency is a function of $\sqrt{T}$.

$$c = \sqrt{gKRT}$$

where

g = acceleration due to gravity

K = adiabatic exponent

R = gas constant

T = absolute temperature of the inlet gas

Since K and R are known and constant in the range of gas turbine engine operation, the frequency becomes a function of $\sqrt{T}$.

Coupler and Frequency Discriminator

To be useful, the pneumatic frequency signal of the temperature sensor must be converted to a usable signal. The pulsing pneumatic signal is converted into an acoustic signal by means of a coupler. In this way, the hot gas of the sensor is not injected into the frequency discrimination system.

The acoustic output of the sensor is fed into a resonator (frequency discriminator) which changes the variable frequency input signal into an acoustic output signal whose amplitude is a function of frequency. The resonator may be a tuned cavity of length equal to one-half the wavelength of the acoustic signal.

Application

Chief advantages of the fluid temperature sensor include rapid response to temperature changes and ability to function in the elevated temperature environment at the turbine inlet. The sensor is constructed of the same materials as those used in the turbine itself.

Weight of the breadboard model temperature sensor without signal conditioning circuitry is about 3/4 pound. The sensor is powered by approximately 1 pound minimum of hot gas withdrawn from the sampling point. Although an ultimate weight reduction of ten to one is possible for certain applications, the second-order time constant of the sensor is degraded at reduced flows. In applications where rapid transient response is important, a weight reduction of four to one appears more reasonable.

With multiple combustor arrangements it may be desirable to sense temperature at several points. In principle, the temperature limiting function of the fuel control can be made to operate in one of two ways: limiting on the mean output of the several sensors, or on the maximum output (hotspot) of any one of the sensors. Considerably fewer logic stages are required to implement the mean temperature limit approach, so it is receiving the most attention at present.

Another approach which permits the use of fewer sensors is to collect the sample gas at several points and read out temperature of the mixed gases on a single sensor. Disadvantages of this approach include (1) the problem of making space for the collection ring near the turbine inlet, (2) the error created by heat transfer and pressure drop along the flow paths, and (3) the slower response to changes in temperature.

SPEED LOOP

The fuel control simulated in the study utilizes a proportional-plus-integral speed control loop. Honeywell is presently conducting a separate study to optimize and implement a proportional speed loop plus a phase lock type of integral loop to provide fast transient response and zero-droop steady-state characteristics, respectively. The proportional loop has the advantage of faster transient response and would therefore be used in conjunction with the phase-lock loop where rapid response is important. For systems in which rapid response is not so important, a reset capability is envisioned to work with the phase detector, allowing zero-droop speed control without the need for a proportional speed sensor.

The proportional (analog) speed sensor has been successfully tested on an engine while the phase-lock speed loop is presently in the analysis and simulation stage of development.

Analog Speed Sensor

Introduction -- Development work during a previous contract on "Fluid Amplifier Turbojet Engine Control Feasibility Studies" resulted in the design of a fluid analog rotational speed sensing device. This device produces a differential pressure output signal which is proportional to the rotational speed of the surface being sensed. A description of this concept follows:

When a flat plate is placed in close proximity to a rotating cylindrical surface, as shown in Figure 60, the hydrodynamic pumping action in the boundary layer between the plate and rotating surface generates a positive pressure ahead of the tangent point and a negative pressure behind the tangent point. Pressure taps placed in the flat plate over the pressure peaks give a differential pressure which is proportional to the rotational speed.

An equation expressing the pressure signal in terms of the variables involved shows that the pressure signal is (1) directly proportional to the dynamic viscosity μ and the angular velocity ω and is (2) inversely proportional to the clearance h between the probe and rotating surface divided by the cylinder (or wheel) radius all raised to the 1.5 power.

$$P_{(max)} = 0.92 \frac{\mu \omega}{\left|\frac{h}{R}\right|^{1.5}}$$

$$\text{at } X = \sqrt{(2/3)hR}$$

where

$P_{(max)}$ = the peak gage pressure or vacuum developed

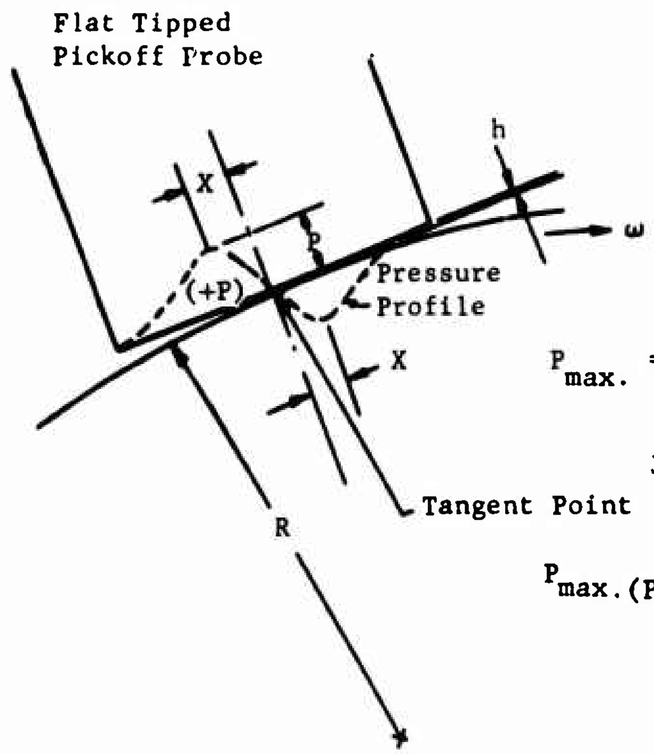
ω = rotational speed - radians per second

μ = fluid dynamic viscosity

h = clearance between wheel and flat surface

R = radius of wheel

X = distance from tangent point to pressure peaks



$$P_{max.} = \frac{3\sqrt{6}}{8} \quad \frac{\omega\mu}{(h/R)^{1.5}}$$

$$X = \sqrt{(2/3)hR}$$

$$P_{max.(Pos)} = P_{max.(Neg)}$$

Figure 60. Schematic of Hydrodynamic Pressure Speed Sensor.

The schematic drawing of this speed pickoff concept shows the variables of the above equation and the superimposed theoretical pressure profile. Distance X is from the tangent point to the positive or negative pressure peak. It is seen from the equation that X is a function only of the clearance h and radius R. By placing pressure taps in the pickoff probe directly over the pressure peaks (that is, at a distance X on either side of the tangent point), a differential pressure can be sensed which is proportional to the angular velocity, ω.

This analog speed sensing device performed well when integrated into the total fluid fuel control system. The feasibility of the device was clearly demonstrated by its simple, rugged design and by its reliable and fast responding output signal. Environmental sensitivity was small over a reasonable range of temperature and pressure. A photo of the analog speed sensor mounted on the test engine in its operating position is shown in Figure 61.

Developments being carried out under AF contract have made significant progress in advancing the design of the analog speed sensor. Three new factors are apropos:

1. New developments have made it possible to increase the air gap by a factor of four to six times. Bearing tolerances are therefore less critical.

2. The analog speed sensor now is capable of operating into an output load without creating significant error due to loading effects.

3. By proper choice of materials, it appears possible to generate a corrected speed signal with the analog speed sensor alone.

The new wraparound analog speed sensor has an output scale factor of

$$\Delta P = \frac{0.945 \mu \omega}{(h/R)^2}.$$

For perfect compensation such that

$$\Delta P = \frac{K_1 N}{\sqrt{\theta}} = \frac{K_2 N}{\sqrt{T}},$$

the factor $\mu/(h/R)^2$ must be a function of T such that

$$\frac{\mu}{(h/R)^2} = \frac{K}{\sqrt{T}}$$

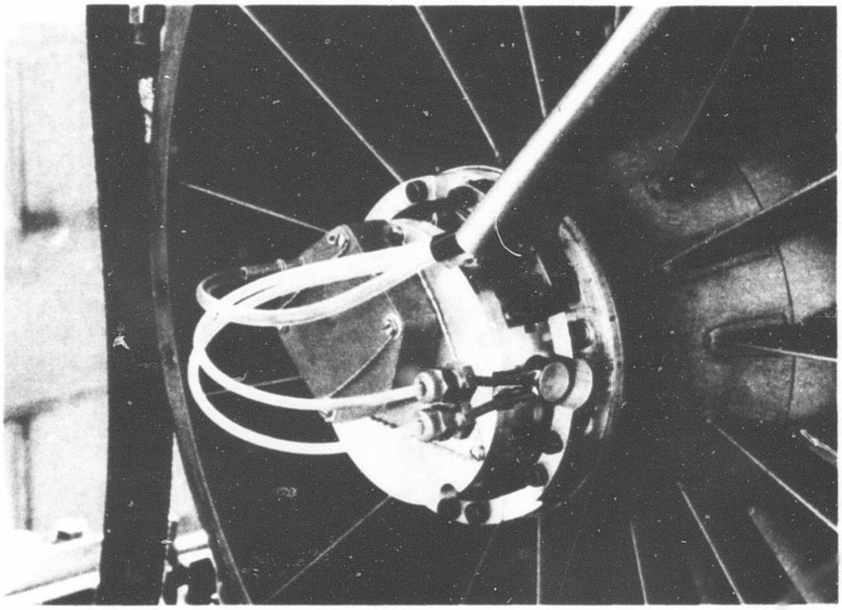


Figure 61. Analog Speed Sensor Mounted on Test Engine.

or $h^2 = \mu\sqrt{T} = h(T)^{1.5}$, assuming $\mu = f(T)$ and $h = f(T)^{0.75}$. A plot of $(T)^{0.75}$ versus T indicates that $(T)^{0.75}$ is nearly a linear function over the range of, say, 400° to 800°R. Thus, the linear coefficient of expansion of parts in the speed sensor nearly approximates the desired speed correction. Using a best straight line, a coefficient of expansion of 3.375×10^{-6} inches /°R yields a maximum error $N/\sqrt{T}$ of 3 percent at the lower end of the temperature range. It is believed that the above coefficient is practical. The use of the analog speed sensor is therefore considered feasible, since, in the speed loop, the analog sensor is used for proportional control during transients and does not influence steady-state speed control.

Fluid Phase-Lock Speed Loop

As described under the Speed Loop heading, a phase-lock type of integral speed loop is proposed to provide zero-droop, steady-state speed characteristics. An abbreviated description of this concept follows.

Circuit Concept -- The concept of a phase-lock speed loop is taken from the electronics field where it is used to control oscillators and is applied to fluids as a potentially attractive concept for engine speed control. The present two-loop proportional speed control takes the form of an analog sensor loop to control rapid changes and a pulsating sensor loop for more accurate steady-state control. By comparison, there are two advantages in using the phase-lock speed control instead of the pulsating speed sensor for the integral portion of the speed control.

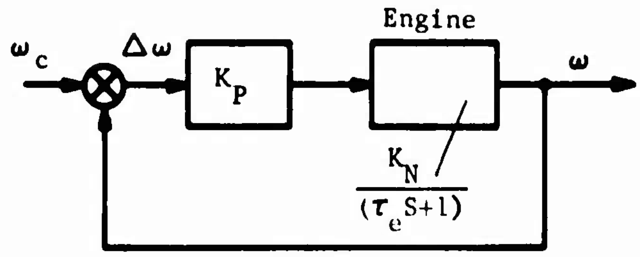
1. There is no steady-state droop.

2. The response is faster.

The disadvantage is that some new (but apparently realizable) components are required for mechanization.

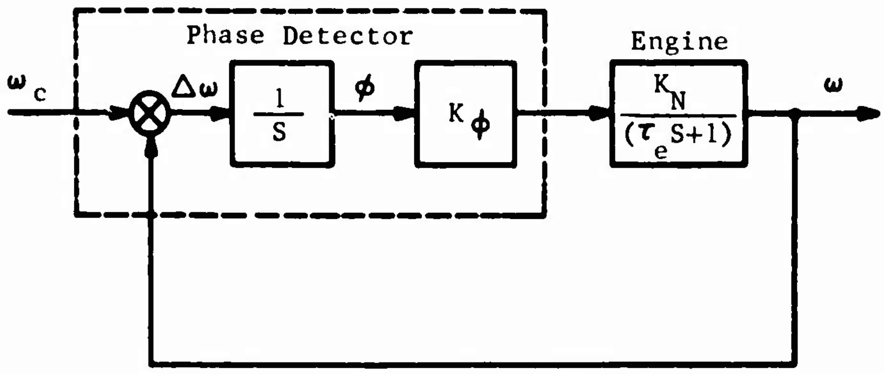
The phase-lock system differs from the usual proportional speed (frequency) control system by virtue of the integration in the forward loop. Proportional control and phase-lock control are depicted in Figure 62. In proportional control, the actuating error ($\Delta\omega$ in the figure) can never go to zero; thus the system has a steady-state droop. The amount of droop can be reduced by increasing the gain but, of course, this approach is limited by the system stability requirements.

On the other hand, a phase-lock system exhibits no steady-state droop because the integration in the loop always drives the actuating error to zero. Thus, the steady-state speed (frequency) is always equal to the commanded value.



Steady-State Error, $\Delta\omega = \frac{1}{1 + K_P K_N}$

A. Proportional System



Steady-State Error, $\Delta\omega_{SS} = 0$

B. Phase-Lock System

Nomenclature

ϕ = Phase angle (rad)

ω = Engine speed (rad/sec)

ω_c = Commanded speed (rad/sec)

K_ϕ = Phase detector gain (psi/rad)

K_p = Proportional gain constant

τ_e = Engine time constant (sec)

S = Laplace transform notation

Figure 62. Proportional and Phase-Lock Systems.

An extremely attractive feature of the phase-lock loop is that no explicit integration process takes place. Instead, the integration is an inherent feature of the detection process. The system utilizes a phase detector (shown in dotted outline in Figure 62) which detects the phase difference between the command signal and the actual speed signal and develops a pressure which is proportional to the phase error. Since phase is the integral of frequency, the integration presented explicitly in Figure 62 is achieved implicitly.

Because the detection process produces an output signal proportional to the sine (or cosine) of the phase angle, the detector output is multi-valued unless phase is restricted to an excursion of $\pm\pi/2$ about the suitable zero of the trigonometric function. This restriction produces a maximum range or excursion in frequency that the phase detector can handle. This lock range, as it is called, is given by L.R. $= \pi K_\phi K_N$. Conversely, if the loop dynamics are fast enough, the phase detector will lock the system (achieve synchronism) to the command signal if the difference frequency does not exceed the lock range.

Referring to Figure 62, it may be seen that the closed loop transfer function is

$$\frac{\omega}{\omega_c} = \frac{\dfrac{K_\phi K_N}{\tau_e}}{S^2 + \dfrac{1}{\tau_e} S + \dfrac{K_\phi K_N}{\tau_e}}$$

This is a second-order system which is never unstable. If there were no other lags or time delays in the system, this equation would actually represent the system, and implementation of the phase-lock concept for jet engine speed control would be simple. One would merely make the loop gain $K_\theta K_N$ high enough so that the lock range would be equal to or greater than the speed range of the engine.

Unfortunately, other dynamics (which for simplicity have been omitted up to this point) are present in an actual engine speed control loop. Consideration of these results in a higher order system, and the loop gain now has an upper bound, beyond which the system becomes unstable. Preliminary analysis indicates that the lock range is much smaller than the engine speed range. For one particular engine, L.R. $= \frac{1}{10}$ (Speed Range).

Since the lock range is restricted, an auxiliary speed loop using the analog speed sensor or some means of moving the lock range along with the engine speed command is required to control large command changes tending to unlock the speed loop.

Where a very fast response is needed, it is advantageous to employ the proportional speed loop in conjunction with the phase-lock loop. In this case, when a large transient change is commanded the phase detector unlocks and the proportional loop controls the engine through the transient. As the new steady-state commanded speed is approached, the phase detector relocks and brings the speed to a zero-droop, steady-state condition.

In other instances where a somewhat slower response is permissible, a reset scheme is envisioned which will permit zero-droop control with the phase-lock speed loop alone, without need for the proportional speed loop.

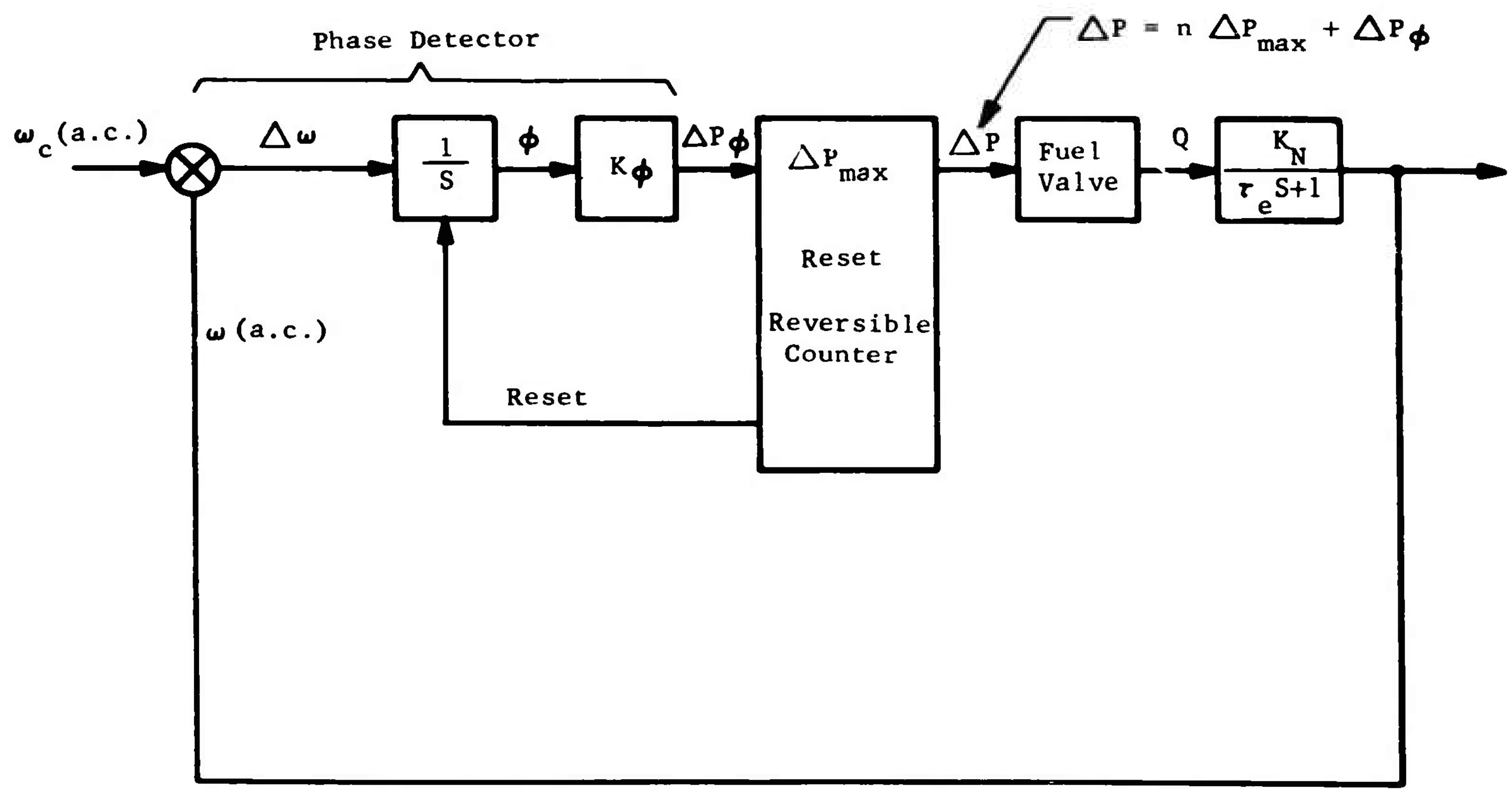
Reset Scheme -- Figure 63 shows a reset scheme which represents the ultimate development of phase-lock technique when the commanded variable is greater than the available lock range. It requires no auxiliary speed control loop and thus represents the simplest arrangement conceptually. If, for any reason, the simple phase-lock control receives an engine speed error signal which so differs from the command signal as to be outside the locking range, the resulting ΔP signal is cyclically unstable, resulting in a sinusoidal variation in engine speed. This unstable mode is prevented by the reset-and-counter feature shown in Figure 63. In order to maintain a ΔP signal suitable for maintaining the engine speed within locking range of the command setpoint, the locking range itself is, in effect, translated along with the engine speed so that the control system never slips out of synchronization. A counter is used to measure the displacement of the locking range and, in time, to pull the locking range (and with it the engine speed) back to the command set point.

Stated another way, when the phase detector is driven to either extreme of the lock range, the "integrator" is "reset" to zero (or some other value - perhaps to the oposite end of the lock range), and, at the same time, a hold circuit clamps the output pressure to the value which corresponds to the lock range extreme. The phase detector is now free to be driven to the lock range again. This is done (n) times by using a counting scheme. The final detector output pressure is then

$$\Delta P = n\Delta P_{max} + \Delta P_{\phi}.$$

Since the process must be reversible, an up-down counting mechanization is required. Quotation marks have been used in the above discussion because the integration action - it will be called - is not explicit, and thus the reset action may be more difficult than with a conventional integrator.

Phase Detector -- The fluid phase detector mechanization adapts the linear mixer concept of the communications field to the fluid regime. The detector output is a pressure proportional to the cosine of the phase angle difference between two sinusoidal signals of equal frequency.



Q = Fuel Flow, lb/sec

ΔP = Fluid Pressure, psi

Figure 63. Speed Control Using Phase-Lock Loop With Reset Concept.

The superheterodyne circuit used in many radio receivers is a common example of a linear mixer. In the radio application, a local oscillator signal (of large magnitude) is mixed with the incoming signal (of much smaller magnitude) and produces a beat frequency which is then processed further to produce the audio output. In adapting this concept for phase detection in fluids, we utilize the same technique, but with a different emphasis. Since the phase detector is intended for use in a phase-lock jet engine speed loop, we mix two signals of the same frequency but which are generally separated in phase. Thus, our aim is to extract the phase angle information from the detector.

The frequencies will be the same when the loop is locked and in steady-state operation. However, the frequencies will be different during response to transients and when the loop is not phase-locked. In this case, the detector output is a beat frequency and is truly a linear mixer in the communications sense.

A proposed mechanization of the fluid phase detector is shown in Figure 64. A reference (local oscillator) signal, $P_c = P_{co} \cos \omega t$, and the engine speed signal, $P = P_o \cos (\omega t + \phi)$, generated by a speed sensor, are summed in a proportional fluid amplifier. The resultant signal is applied to a fluid diode which acts as a detector. A suitable filter placed at the output of the diode will remove the carrier frequency, leaving only the envelope.

The resultant pressure signal is

$$P_{\phi} = K(k_f - k_r)P_o \cos \phi$$

where

k_f = forward slope of diode

k_r = reverse slope of diode

Implicit in phase detection techniques is the limitation in the excursion of phase angle to $\pm \pi/2$ radians. When the phase detector is used in a closed loop, the system is said to be in lock if the phase angle remains within these limits. Excursion beyond this range causes the loop to unlock, resulting in a beat frequency output given by

$$P_{\phi_{unlocked}} = K\ (k_f - k_r)\ P_o \cos\ (\omega_c - \omega)\ t;$$

hence the need for the reset circuit described above.

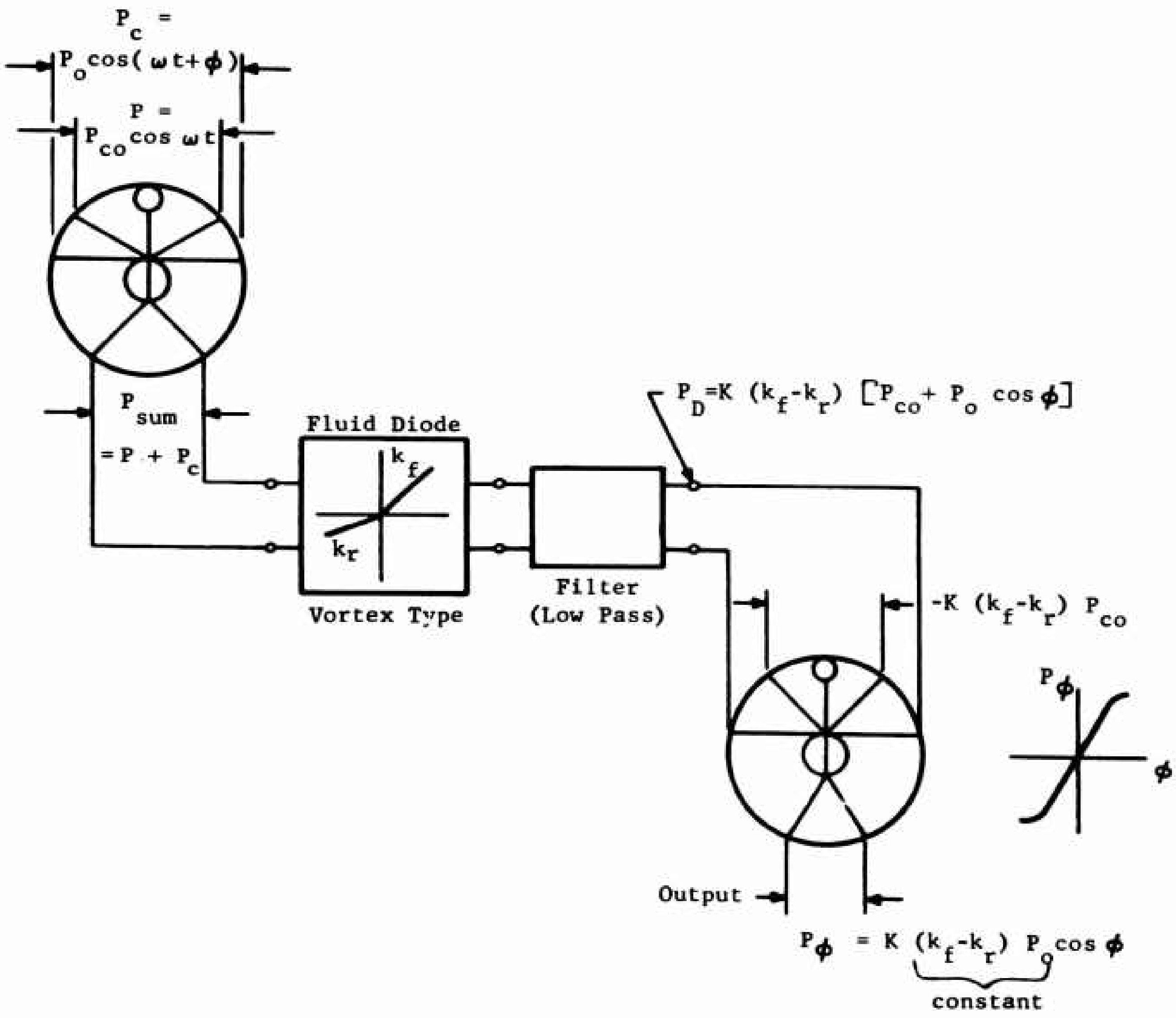


Figure 64. Fluid Phase Detector.

The fluid phase-lock speed loop concept with fluid phase detector will be verified experimentally in the Honeywell laboratory in the near future. For this most attractive method of applying the phase-lock concept, success will depend on several factors among which the following are most significant:

1. Availability of a fluid diode with significantly differing front and back slopes.

2. Propagation of the carrier frequency through the detector without distortion.

3. Mechanization of a reset/counter device.

Should the reset/counter mechanization prove too difficult, other phase-lock application schemes are possible, such as employing an auxiliary (analog) speed loop together with switching logic for selection of the phase-lock or analog mode.

The outstanding feature of speed loops utilizing the phase-lock principle is their ability to control speed with zero droop.

Corrected Engine Speed

In the START and RELIGHT CONTROL as well as in the basic fuel control, there may be need for an engine speed signal which is corrected for variations in compressor inlet total temperature T_{2T}, particularly where the engine is to be started at elevated field altitudes or in flight. This signal, $N/\sqrt{\theta}$, is shown in Figures 40, 57, and 59 where N = RPM, and $\theta = T_{2T}/520°R$. It may be possible to obtain a corrected speed signal directly from the analog speed sensor. Another approach for computing the square root temperature correction to engine speed uses a pulse speed sensor as follows:

Pulse (Integral) Speed Sensor -- In the system depicted in Figure 59 a pulsating type speed sensor provides the corrected speed signal. The envisioned method of operation involves the generation of pressure pulses by means of a rotating, perforated disc (reference Figure 59). Pulse frequency (f) is proportional to the engine speed (N). Each positive pulse turns on the bistable amplifier, followed by a negative pulse in the opposite control port which cuts off the amplifier. Pulse width (w) of the amplifier output signal is determined by the sonic velocity (c) through the delay line (ℓ). But the sonic velocity is related to $\sqrt{T}$ where T is the absolute gas temperature in the delay line.

$$\omega = \frac{l}{c} = \frac{l}{K\sqrt{T}} = \frac{K_2}{\sqrt{T}}$$

The integrated output signal (S) is

$$S = \frac{p\ f\ K_2}{\sqrt{T}} = \frac{K_2\ N}{\sqrt{T}}$$

where

p = pulse pressure (i.e., pulse height)

$f \propto M$

Thus, the $\sqrt{T}$ function is obtained through variation in pulse width and the pulse frequency is proportional to N, resulting in a corrected speed signal.

Tests are presently being undertaken on another program to determine the feasibility of this approach to computing corrected speed.

Frequency Divider -- One problem in using the pulse speed sensor to compute corrected speed is that it is limited to a frequency response of about 200 cps (for a dwell of 10 times the pulse width of 0.5 millisecond). Since bistable elements have higher frequency response, one solution is to employ them as frequency dividers (i.e., a fluid gear reduction train) to reduce the speed signal frequency to a rate which the pulse speed sensor can handle.

Design speed for the study engine is 54,000 rpm or 900 cps. A three-element counter circuit will reduce this frequency by 1/2 x 1/2 x 1/2 = 1/8 or 112.5 cps, as shown in the block diagram of Figure 65. The purpose of the differentiator is to make spikes out of wide pulses, permitting increased strength in feedback and thus higher frequency response, without causing oscillation.

Figure 65 illustrates the old method of counter switching which depended on the control "bias" set up by convection entrained in the control loop. This approach was frequency-limited. Short feedback loops now accomplish bias preparation for the following pulse to much higher frequency limits. The schematic diagram of a counter bit circuit developed by Honeywell is shown in Figure 6 5. A pulse signal input causes an output at (1) and a feedback to control port (b). By venting this feedback near the control port, the feedback signal is weakened so that it cannot switch an established flow. Once the inlet flow is removed, however, this feedback signal does cause the next following input pulse to switch the output signal to outlet (2). In this manner, two input pulses result in only one output pulse at outlet (1), making a frequency divider.

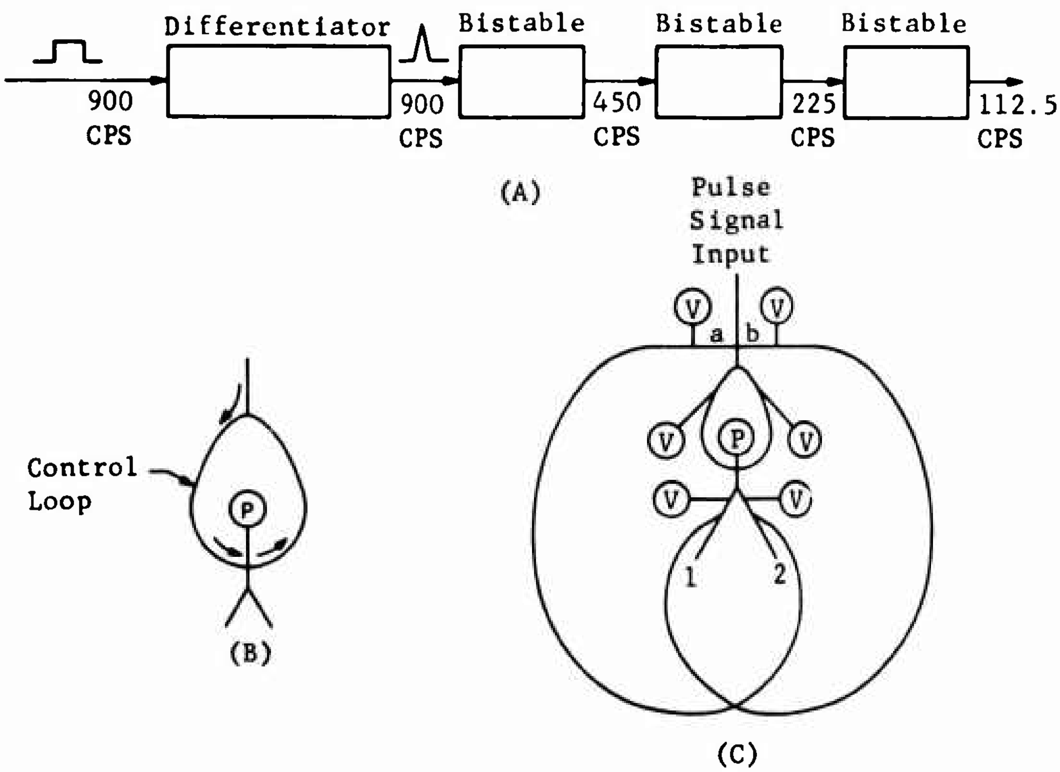


Figure 65. Block Diagram and Schematic of a Frequency Divider.

FLAMEOUT MONITOR

Through work sponsored by USAAVLABS, contract DA 44-177-AMC-305(T), Honeywell has demonstrated that it is possible to develop a flameout monitor to respond to loss of flame within 0.03 second. Two system approaches, one fluidic and one electronic, have been breadboarded and successfully tested on a turbojet engine. The fluidic system senses the temperature of the hot gas and determines whether or not the temperature is sufficiently high for combustion. The electronic system actually "looks" at the flame with a flame sensitive power tube to determine if flame is present.

Electronic Flameout Monitor

In this approach, a flame sensitive power tube "looks" in at the burner flame through a quartz window. When a flameout occurs, an output signal is produced which can be utilized to energize a cockpit warning light or the ignition logic circuit. The battery operated signal conditioning circuitry is contained in a small can, shown together with the other components in Figure 66.

Fluidic Flameout Monitor

The fluidic flameout monitor is built around the fluid temperature sensor described previously. The fluidic system does not react to startup in the same way as the electronic system, because it takes a certain amount of time to build up pressure in both the burner and the fluid temperature sensor. This is not important for the flameout application but may be a factor in the start sequencing application.

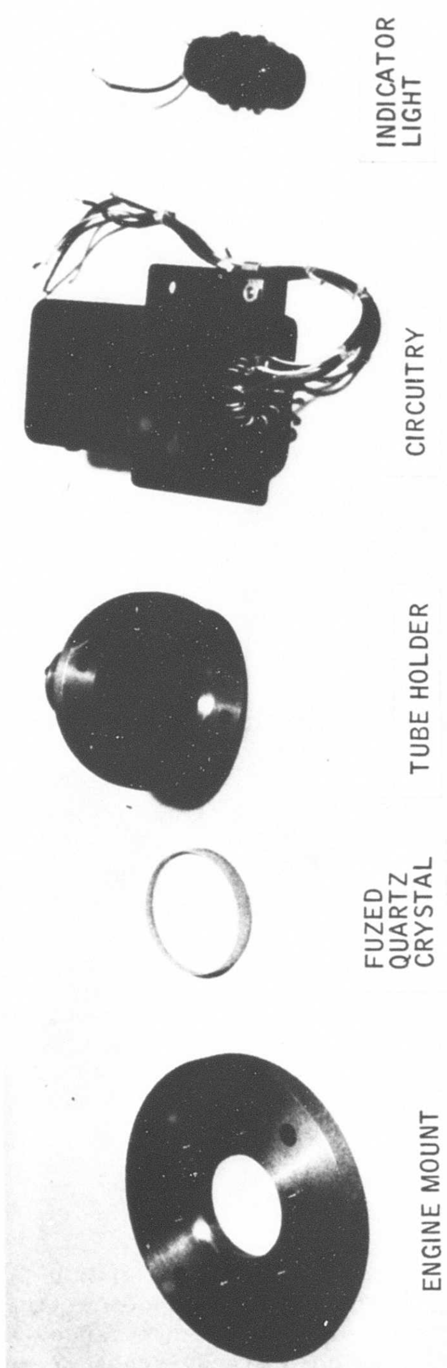
FLUID AMPLIFIERS

Selection and implementation of bistable logic elements have been treated under the section entitled START AND RELIGHT SEQUENCE CONTROL, page 126. The fuel control system will employ proportional and summing amplifiers as well as bistable elements. A basic description of all three types is given in Annex B, Fluid Amplifiers.

COCKPIT SPEED CONTROLLER

The cockpit speed controller, or throttle, for use with a fluid fuel control system is a pilot-operated device which produces a linear differential pressure output versus input position. The selected speed signal is summed with the actual speed signal to provide a steady-state error signal which controls the engine fuel flow.

The linear fluid potentiometer is shown schematically in Figure 67.


INDICATOR LIGHT
CIRCUITRY
TUBE HOLDER
FUZED QUARTZ CRYSTAL
ENGINE MOUNT

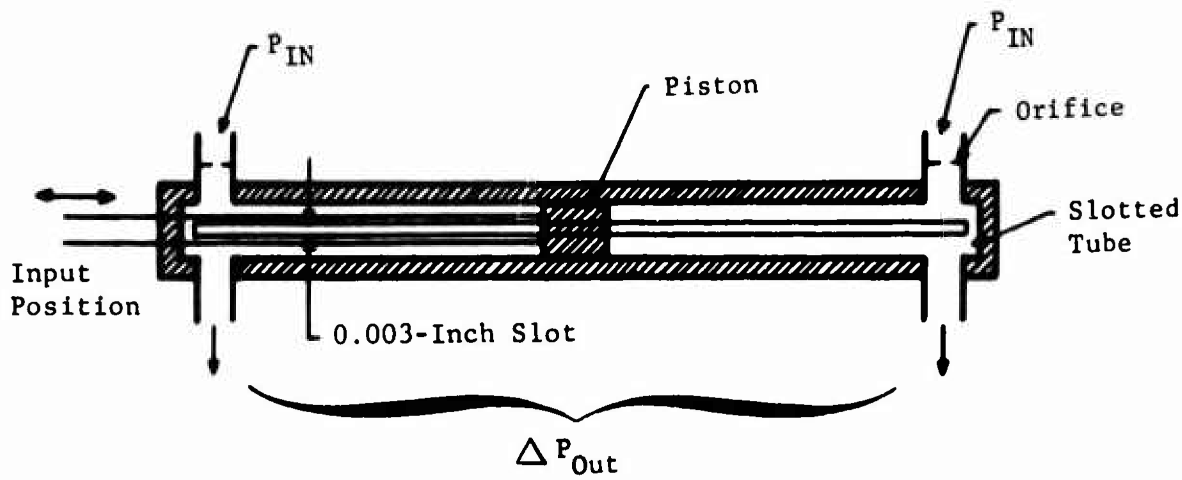


Figure 67. Linear Fluid Potentiometer.

The operation of the linear fluid potentiometer is as follows: P_{in} is established as a logical function of the pressure available for overall system operation. With the piston located halfway between the output taps, the ΔP_{out} is zero. The input orifices are sized with the established P_{in} and with the output taps connected to the device they will operate. Changing the orifice size adjusts the output level of the potentiometer to match correctly the input impedance of the device to be controlled. A good output level is one-half of P_{in}. The output pressure differential can be varied by moving the piston back and forth in its slotted cylinder, which varies the amount of leakage from the cylinder and establishes the pressure on each side of the piston.

A photo of a throttle with the fluid linear potentiometer mounted in a quadrant is shown in Figure 68. This throttle has been used as part of a fluid fuel control system to operate a J85 turbojet engine. In actual practice, the fluid potentiometer would probably be located with the fuel control on the engine. The fluid potentiometer would then be operated by means of a cable or linkage connected to the throttle lever.

FUEL METERING VALVE

Requirement

Output of the fuel control system is metered fuel. Thus there is need for a device which converts a pneumatic control signal to fuel flow rate. Two approaches have been investigated. One is a pure fluid device which shows promise but needs considerable research work before it can be applied. The second is a more conver 'onal fluid pressure-to-mechanical position device which is filling our ˙ ˍdiate needs for system development hardware.

The pure fluid device is a two-fluid vortex chamber. Photographs and a technical description of this device are given in Annex B, entitled TWO-FLUID VORTEX CHAMBER FOR FUEL METERING.

Pneumohydraulic Flow Control Transducer

The need for further research on the vortex chamber, plus the need to return fuel from the vortex chamber to a fuel reservoir maintained at ambient pressure, has led to choice of a more conventional method for controlling fuel flow on current engine control development programs. While ultimate development of a pure fluid fuel control valve is desirable from the standpoint of no moving parts and less trouble from fuel contamination, a pneumohydraulic transducer possesses superior frequency response, having a second-order response characteristic frequency of 10 cps minimum at -6 db compared to 1.2-cps bandwidth for first experimental vortex chambers. The pneumohydraulic transducer should not suffer from high temperature materials problems, since the fuel provides a convenient heat sink.

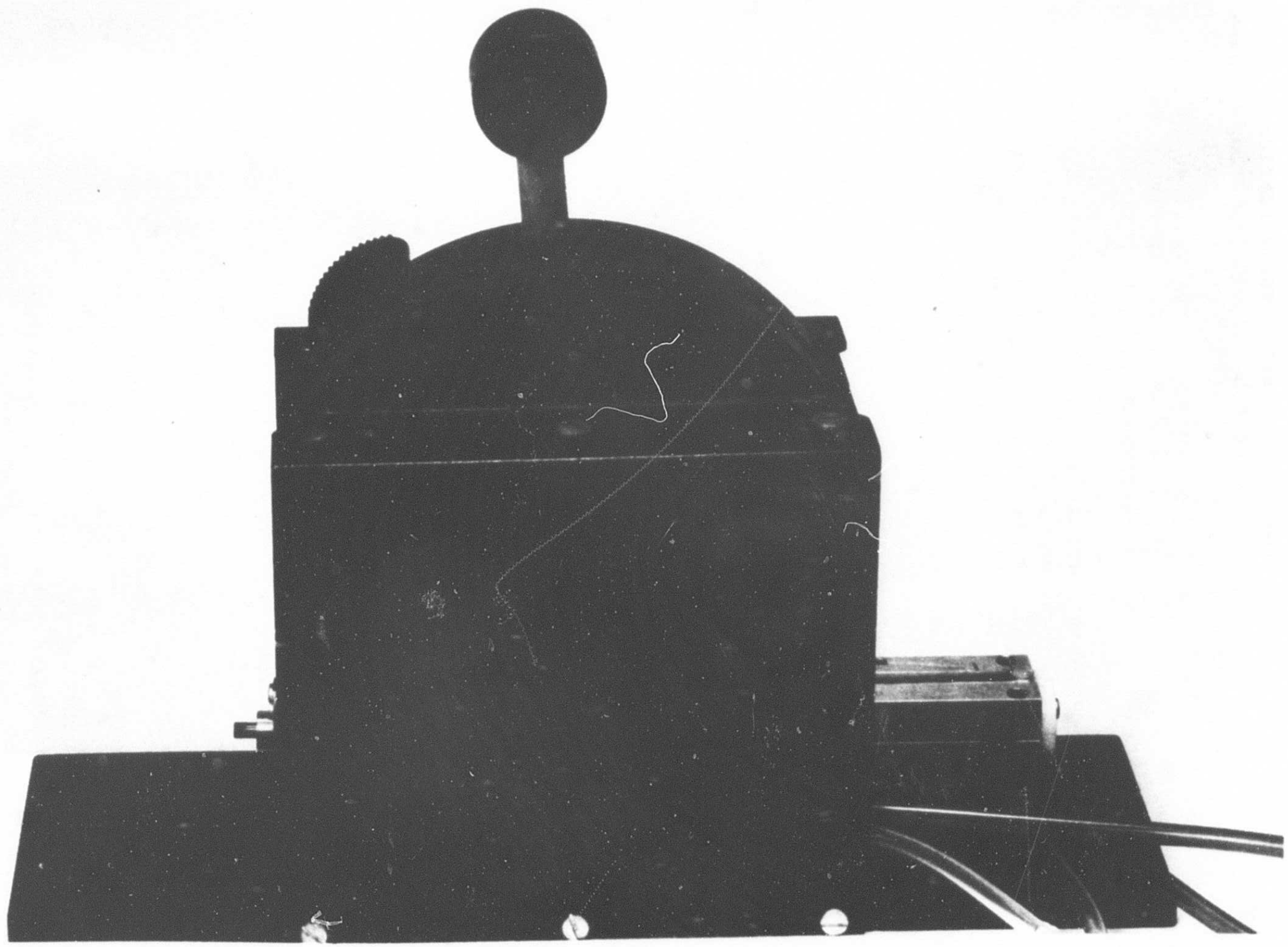


Figure 68. Photograph of a Fluid Throttle Lever.

Figure 69 shows the hydraulic schematic of a servo valve fabricated to Honeywell specifications by Hydraulic Research and Manufacturing Company. This device is a modified version of the Model 263 Electro-Hydraulic Flow Control Transducer in which the torquer coil has been replaced by a pair of bellows actuators. Gain characteristics are shown in Figure 70. Internal leakage at 0-percent flow is approximately 0.15 gallon per minute.

Present experimental models are typically designed to furnish approximately zero fuel flow at -3-inches Hg, ΔP pneumatic input and 11 gallons per minute (with 175-psi fuel ΔP) at +3-inches Hg input. An outline drawing is given in Figure 71. Weight is approximately 1 pound.

The disparity between the idealized linear flow curve and the actual flow test curve may be due to flow effects associated with the spool. The effect of transducer flow nonlinearity on the engine control system is to require a loop gain reduction to assure stability at the steepest portion of the flow curve. This reduction in system gain results in a somewhat slower system time response. Indications are, however, that careful valve design can greatly reduce or eliminate the flow nonlinearity.

Emergency Mode

Provisions for clamping on the spool to provide direct mechanical fuel control were described under MANUAL OVERRIDE PROVISIONS. In addition, some sort of adjustable mechanical stop or emergency pressure bias is needed to protect against loss of CDP pressure to the fluidic power supply. If the fuel metering valve has the characteristics shown in Figure 70, complete loss of control input pressure means a zero ΔP input signal resulting in a 50-percent fuel flow. Provisions need to be made to protect against such a potential overspeed condition.

Series Versus Bypass Installation

Current thinking favors utilizing the fuel metering valve in the series system configuration. Factors involved in choosing between bypass on series fuel control are briefly as follows:

BYPASS FUEL VALVE

Advantages

1. Simpler metering scheme
2. Throttle and ΔP relationship more linear

Disadvantages

1. Instability due to positive feedback from the fuel pump
2. Stop scheduling required
3. Inherently complex
4. Gain scheduling more complex than for series valve

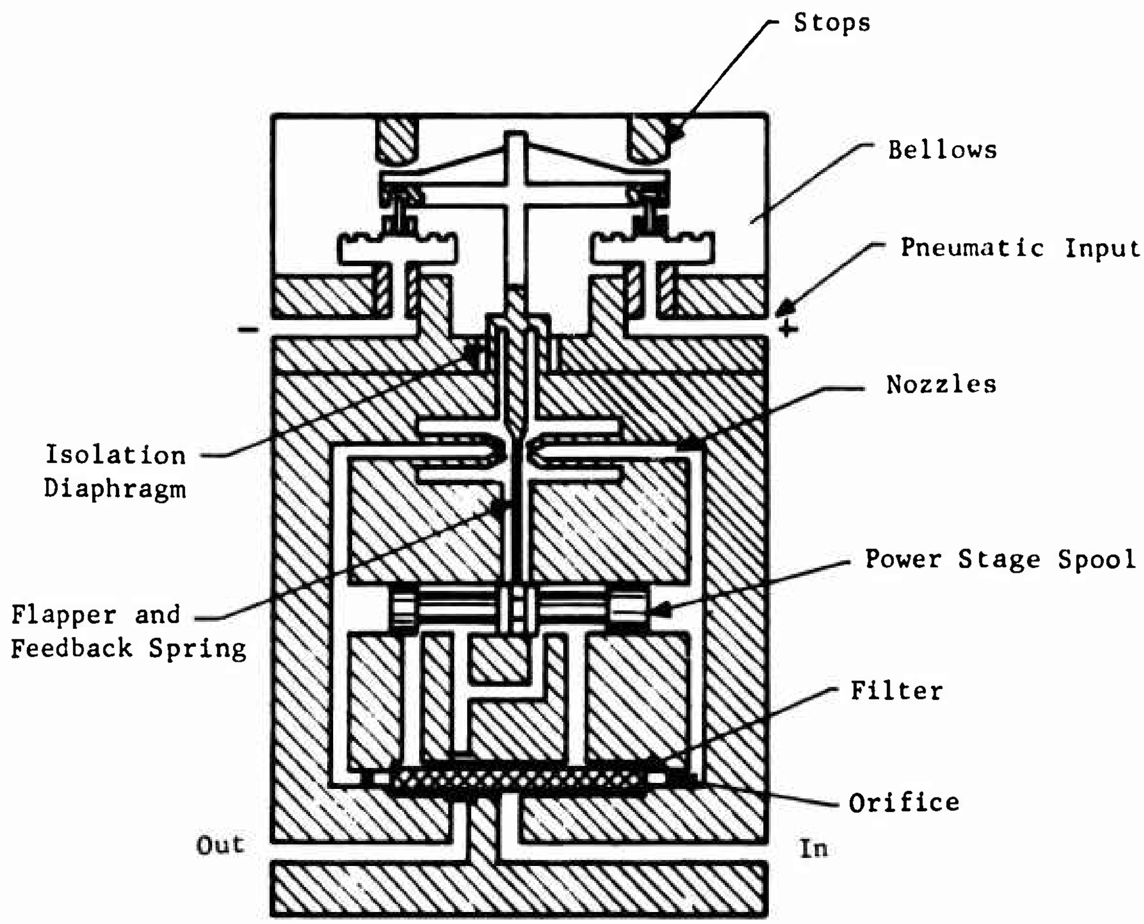


Figure 69. Hydraulic Schematic - Pneumohydraulic Flow Control Transducer.

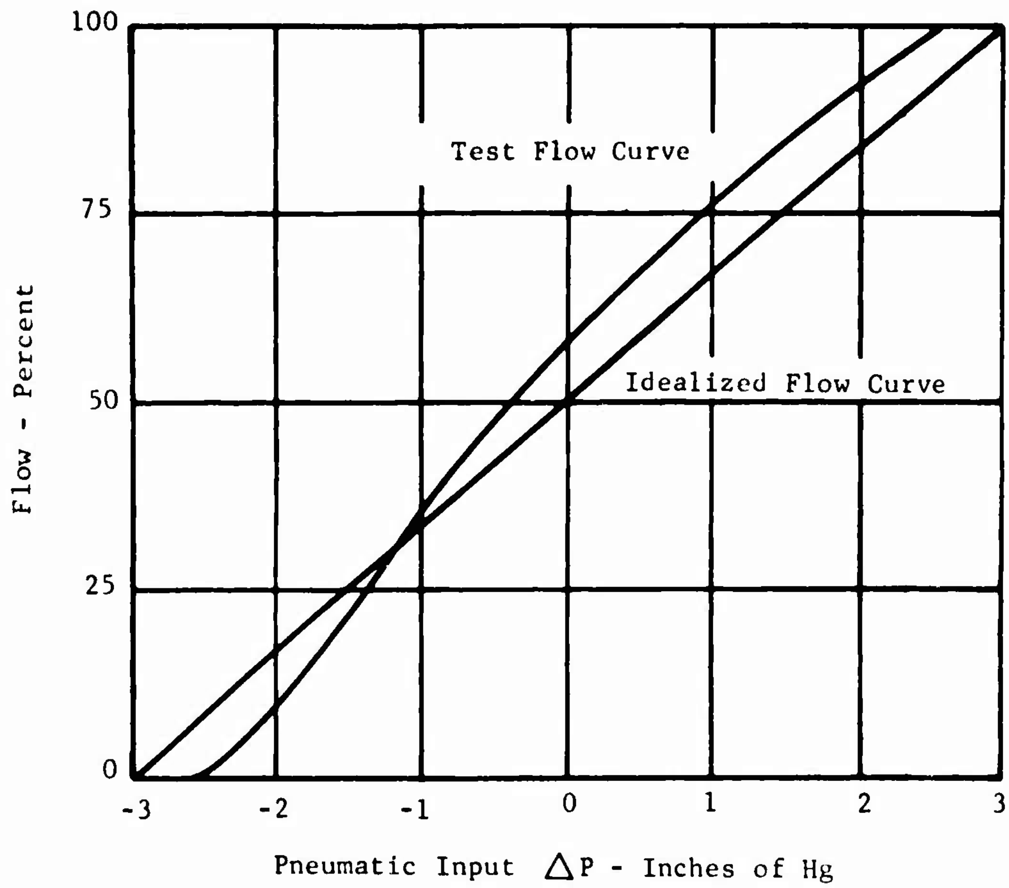


Figure 70. Flow Gain Curve - Pneumohydraulic Transducer.

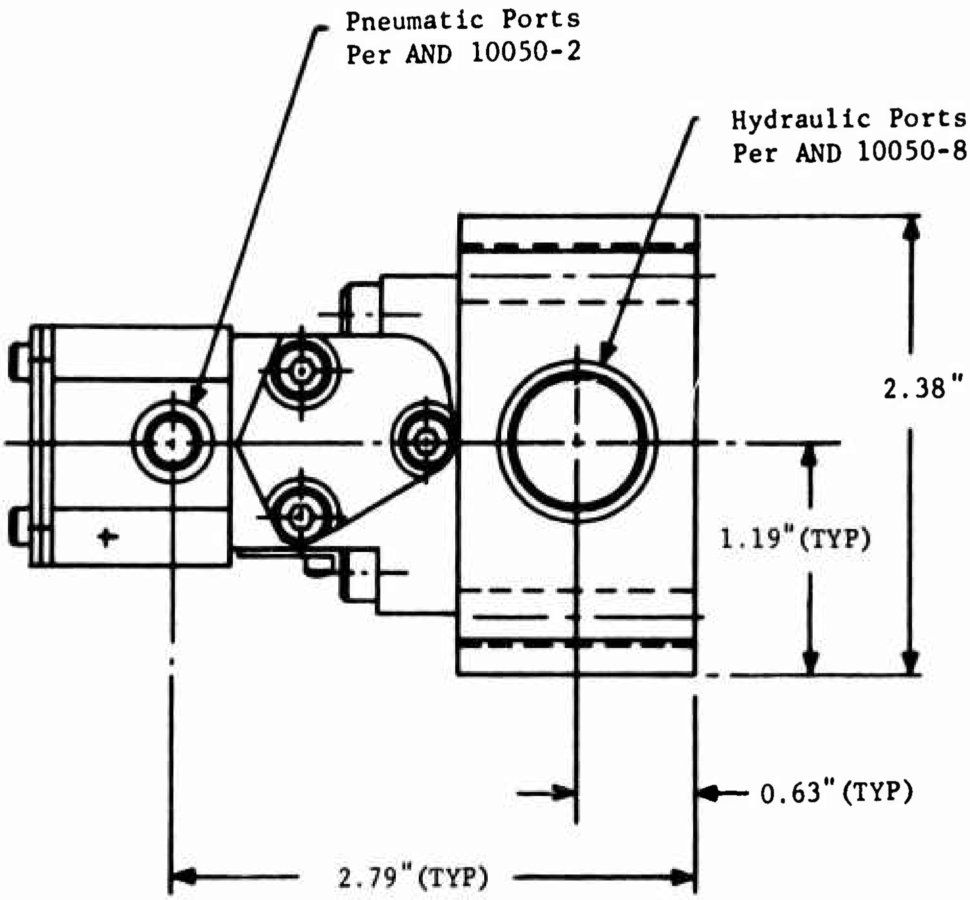


Figure 71. Installation Drawing of Pneumo-hydraulic Transducer (1 of 2).

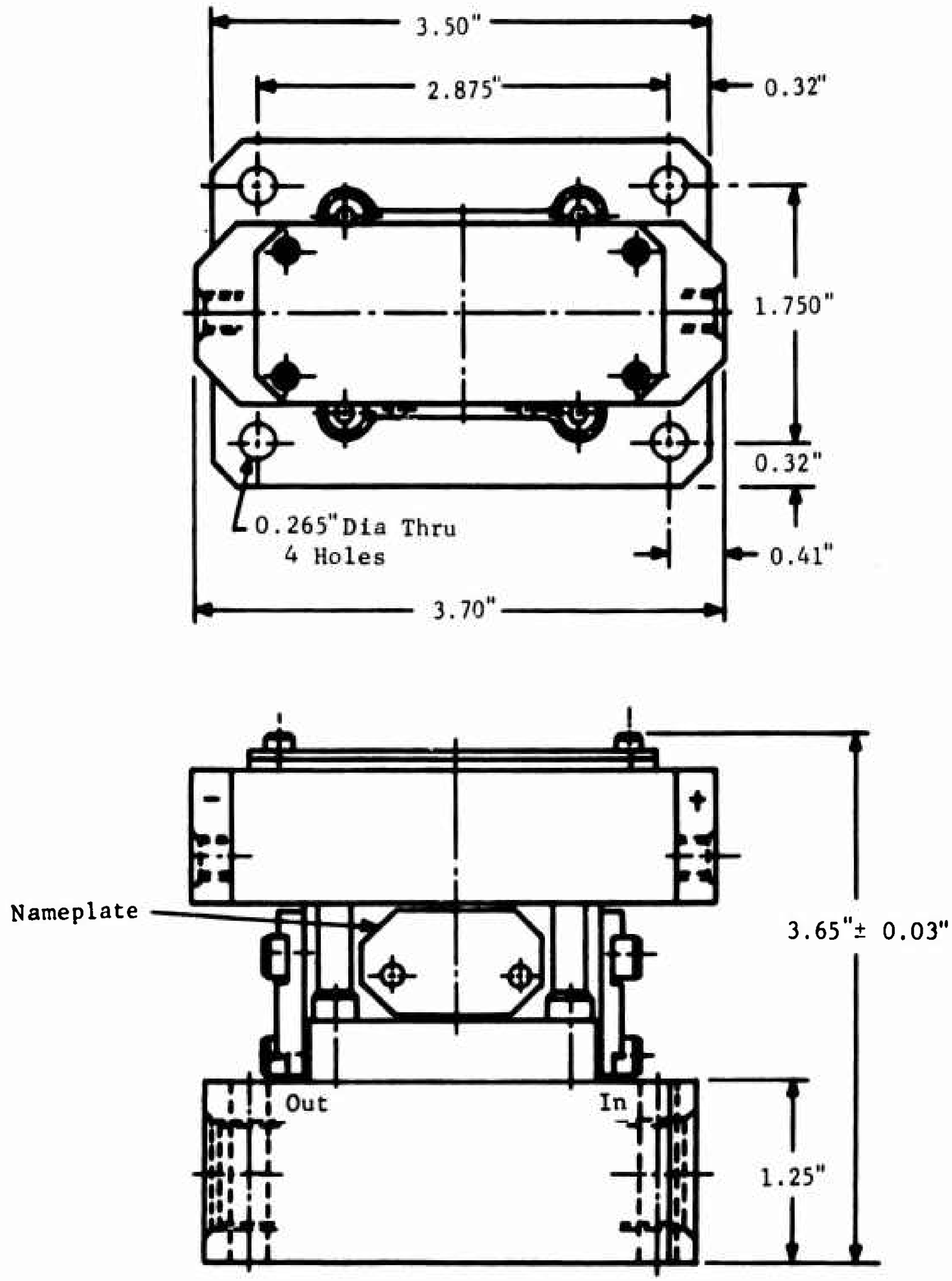


Figure 71. Installation Drawing of Pneumo-hydraulic Transducer (2 of 2).

SERIES FUEL VALVE

Advantages

1. Positive response

Disadvantages

1. Throttle compensation necessary
2. Metering system mechanically more complex (regulator needed to maintain constant fuel pressure at the metering valve)

The following specification for the flow transducer is typical although it is specifically applicable to the bypass configuration. In order to make a first approximation as to the required flow capacity, a direct-driven, positive-displacement fuel pump is assumed, having a flow discharge double that required by a 4.35-pound-per-second W_a engine.

Preliminary Specification for Pneumohydraulic Flow Control Transducer

Scope - This specification defines the requirements necessary to provide a bypass fuel valve for controlling fuel flow to a small gas generator.

Applicable Specification - MIL-E-5272C

Control Medium - Compressed air

Hydraulic Medium - JP4 and contamination passed by filters per AF SCM 80-1

Flow Gain - See Figure 70.
100-percent flow = 1.4 gallons per minute with 175-psi pressure drop

Internal Leakage at 0-Percent Flow - 0.03 to 0.05 gallon per minute

Weight - Less than 0.5 lb

Pressure Level of Pneumatic Input - 4 inches Hg gage

ΔP Pneumatic Range - ±3 inches Hg

Maximum Fuel Pressure - 650 psi

Frequency Response - Based on a second-order response, the output of the valve shall not be down more than 6 db at 10 cps.

Vibration - Per MIL-E-5272C, Procedure XII

Temperature - The valve will be required to operate at ambient temperatures of -65°F to 300°F. Temperature of the pneumatic medium may reach 850°F at the source. Due to the dead-ended nature of the pneumatic input circuit, however, the maximum temperature of the pneumatic medium at the valve is expected to be under 300°F.

Quality Assurance Provisions - To be determined

Reliability - 10,000 hours MTBF

FUNCTION GENERATORS

For surge protection, a set temperature generator is needed in the fuel control to compute the allowable turbine inlet temperature as a function of corrected speed. Another function generator is required in the start and relight sequence control to provide fuel flow rate as a function of speed or corrected speed during starting. A program is under way, under Air Force contract, to investigate the use of a fluid function generator to meet these requirements. This device is basically a proportional amplifier with a specially shaped splitter.

PACKAGING AND INSTALLATION

PACKAGING FLUID ELEMENTS

Packaging is the task of assembling components into systems or subsystems to comply with specified requirements. Packaging objectives generally must satisfy such demands as minimum volume, weight, cost or environmental protection. These demands are not necessarily compatible. Fluid and electronic component packaging are similar in many respects; therefore, previous guidelines established for electronic packaging are helpful.

The dominant packaging considerations include:

1. Physical size - concerns the number of individual components which the package will house
2. Environment - the pressure, temperature, vibration, etc., the unit must withstand
3. Repairability - repair or replacement of systems and subsystems, accessibility of the higher maintenance items
4. Production - the adaptability of the package to acceptable fabrication techniques
5. Control Rigging - accessibility of adjustments for throttle calibration, etc.
6. Economics - cost considerations
7. Weight - overall system weight resulting from various packaging approaches

Fluid elements have inherent qualities which lend themselves particularly well to packaging. The logic elements are simple and rugged. They can absorb large shock and acceleration forces and high temperatures without damage. There are also problems associated with fluid element packaging. Impedance matching is the most serious of these.

For a fluid system application, impedance matching is the task of coupling fluid elements while maintaining their characteristics and obtaining the required pressure and/or flow output. Interrelation between coupled fluid elements presently must be determined experimentally, although Honeywell is progressing with analytical approaches. Consequently, testing of coupling effects in stages is necessary before the final packaging.

Packaging of the pneumatic system can be accomplished in many different configurations, depending upon the engine, its application, and requirements of the fluid system itself. The environmental properties of fluids allow components to be hard-mounted anywhere inside or outside the engine. However, maximum response and good signal strength require that distances between sensors and logic be kept to a minimum. Optimum performance and minimum weight dictate that sensors be located at the source of measurements inside the engine, perhaps integrated into the structural members of the engine, rather than transmitting the condition to sensors outside the engine. Unless the sensors are integrated during initial design of the engine, however, this may prove impractical, and externally mounted sensors and logic may be required.

Two basic techniques are available for interconnecting fluid logic elements: the "in-line" (or "strung out") and the "stacked" methods. For the in-line method an entire system or subsystem consisting of several groups or blocks of interconnected fluid elements is etched, cast, or otherwise processed into a single plate. The interconnections are relatively long and great in volume, but the flow path is generally smooth and free of abrupt changes in direction. The in-line method probably will take up more packaging volume than the stacked method but lends itself well to subsystem modularization.

In the stacked method, plates incorporating a single stage or a small grouping of stages are laminated one on top of the next to form a functional block which is mounted to a manifold plate. Generally, interconnections are made by through-passages formed into each element. This method lends itself to dense packaging as well as to easy replacement of individual stages during development or maintenance. Development is simplified due to the ease of selection and changing of the aspect ratio (ratio of depth to width of the power nozzle) of individual stages for impedance matching purposes. The stacking method will probably be the easiest to equip with test points for checkout and maintenance. The volume of interconnections between stacked elements is small, but the flow path must make a 180-degree bend after every element. The stacked method of packaging is illustrated in Figures 72, 73 and 74.

APPROACH TO PACKAGING THE SYSTEM

Honeywell has expended considerable effort towards special fabrication and packaging of fluid elements and systems. The materials and fabrication effort accomplished under the initial Air Force engine control study was concerned with special high temperature materials and techniques - not packaging per se. Follow-on work for the Air Force is aimed at developing concepts for solution of the most critical packaging problems which would be obstacles to future application. This work is currently in progress and, together with the experience gained in building several demonstrator speed and temperature loops, has served to extend packaging concepts along two general lines:

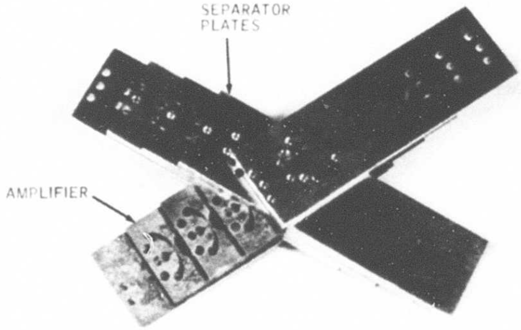
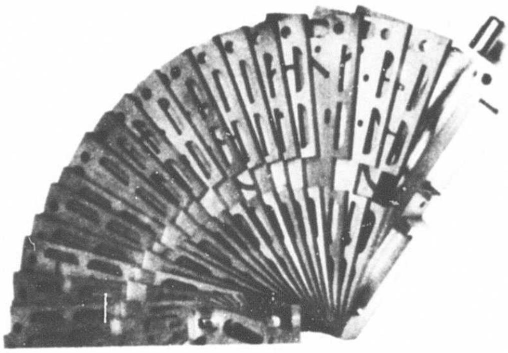
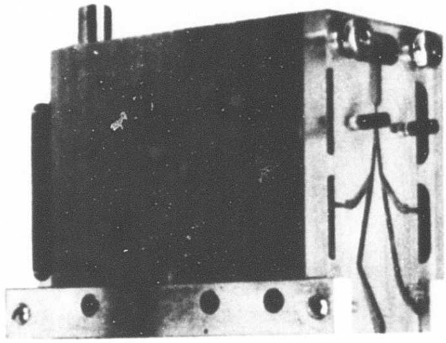


Figure 72. Direct Coupled Proportional Amplifier Cascade.



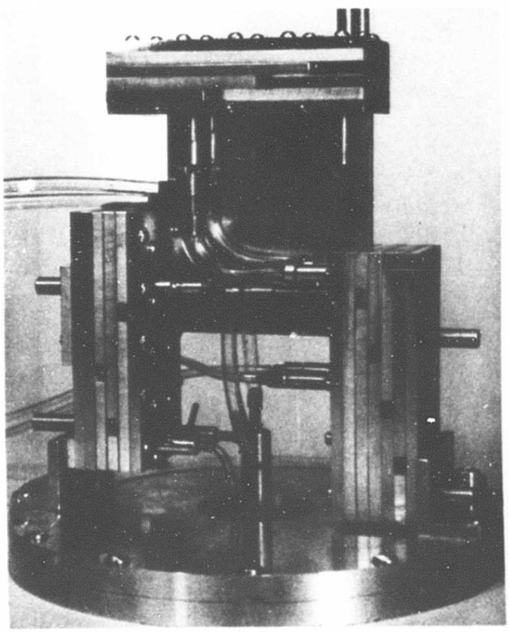
(A) Exposed View



(B) Assembled View

Figure 73. Packaged Fluid Logic Elements.

Figure 74. Packaged Fluid System With Rate Sensor and 11 Cascaded Elements.



1. Case I Sensors integral with engine - logic and controls external

2. Case II Sensors and logic in a common external package, insofar as possible

Case I - Sensors Integral With Engine

This type installation is expected to predominate. Even sensors which are not integrated into the engine will be mounted as close to their probe sources as possible and, like integral sensors, will require transmission of signals to a central control package. The following discussion illustrates areas which would be subjects of investigation in working out a fuel control package for a specific engine.

Analog Speed Sensor -- The analog speed sensor could be mounted inside the engine and measure boundary layer pressures on the main bearing shaft or on specially machined surfaces of the compressor rotor. The best locations in the engine are those where maximum ambient pressures occur and temperatures of the metal structure are relatively constant. Practical problems are: shaft runout, installation provisions, ease of replacement or cleanout, dust or oil contamination, and output signal transmission. Some of the associated problems, such as the effect of ambient variations, are being considered in the basic sensor development. The packaging problems which would be considered are concerned with the physical practicability of the installation and innate contamination qualities which may preclude internal mountings.

Digital Speed Sensor -- There appears to be no advantage to an integral mounting unless compressor blade passage or some other inherent means of pulse generation occurs. Assuming this were possible, the critical problem would be operating the loop at a higher frequency than is now used.

Temperature Sensor -- Adequate safety margins will probably require that turbine inlet temperature be measured at several locations in a cross section of the duct. This could be accomplished simplest by containing the sensor cavity, or at least the total head probe passages, within a turbine stator blade. The thin stator blade cross section would require that consideration be given to: (1) the effect of shim-like thickness of the cavity (low channel depth to width ratio) on operation of the sensor, (2) the effect upon response due to body heating of the cavity air, (3) structural weakening of stator blade, and (4) interference with blade cooling.

An alternate plan might be to mount the sensor as close to the turbine shroud casing as possible and duct hot gases through stator blade probe passages or special total pressure probes or stator pressure taps. If adequate pressure ratios exist between the stream and ambient exhaust pressures, static taps may suffice to power the sensor and thus simplify the installation. Otherwise, total head probes inside the engine are necessary. Signal transmission and decoupling requirements are additional problems critical to the specific installatior.

Logic and Controls -- There are no problems in logic packaging which Honeywell is not now investigating on company-funded projects and other fluid contracts. The problems of transmitting sensor signals long distances through tubing which has thermal gradients, bends, or other discontinuities must be considered as they occur in the specific engine installation. The type of interconnection or signal coupling device must also be considered for ease of disassembly, inspection, preventative maintenance or repair and rigging. Regardless of the control's reliability, military procedures will require planned periodic inspections.

Case II - External Packaging

This case is equivalent to conventional fuel control installations. An exploded view showing how a typical external package might appear is shown in Figure 75. This approach avoids the contamination, shaft runout and environmental problems associated with integral installation of the analog speed sensor in the engine. In this case the speed sensor is integrated into an external package driven by direct coupling to the engine rotor. An analog speed pickoff is mounted at the periphery of the wheel. A pulsating (integral) speed pickoff incorporated in the same assembly senses pulses of air passing through a hole in the rotating wheel.

The fluid logic is arranged in modular fashion using the in-line method. Modules are laminated together, each module being functionally complete within itself. Typical modules include the analog speed amplifier cascade and summing amplifier, reference oscillator (for pulse speed loop), pulse speed comparator circuit, and pulse speed proportional amplifier cascade and summer. The first stages of signal conditioning after the temperature sensors will have to be located adjacent to the sensors. Later states, including the temperature limiting circuit, could be modularized and incorporated in the external package.

The external package could also contain the fuel pump and fuel metering valve. In a conventionally arranged compressor, the control could be incorporated into the bullet nose, with pneumatic and fuel connections made through a strut. Aerodynamic considerations will in some cases limit the size of the strut and therefore influence the number of fluid lines which can be connected to such a package.

The outer case shown in the illustration is included for mechanical protection only. It is not needed for pressure sealing and could be deleted if necessary in favor of easier access to the modules for checkout and control rigging.

The external package approach would permit compact packaging, ease of disassembly, and flexibility for the addition or subtraction of various loops. Critical problems common with other packaging methods include the interconnections necessary between integral sensors and the package and the required sealing between modules.

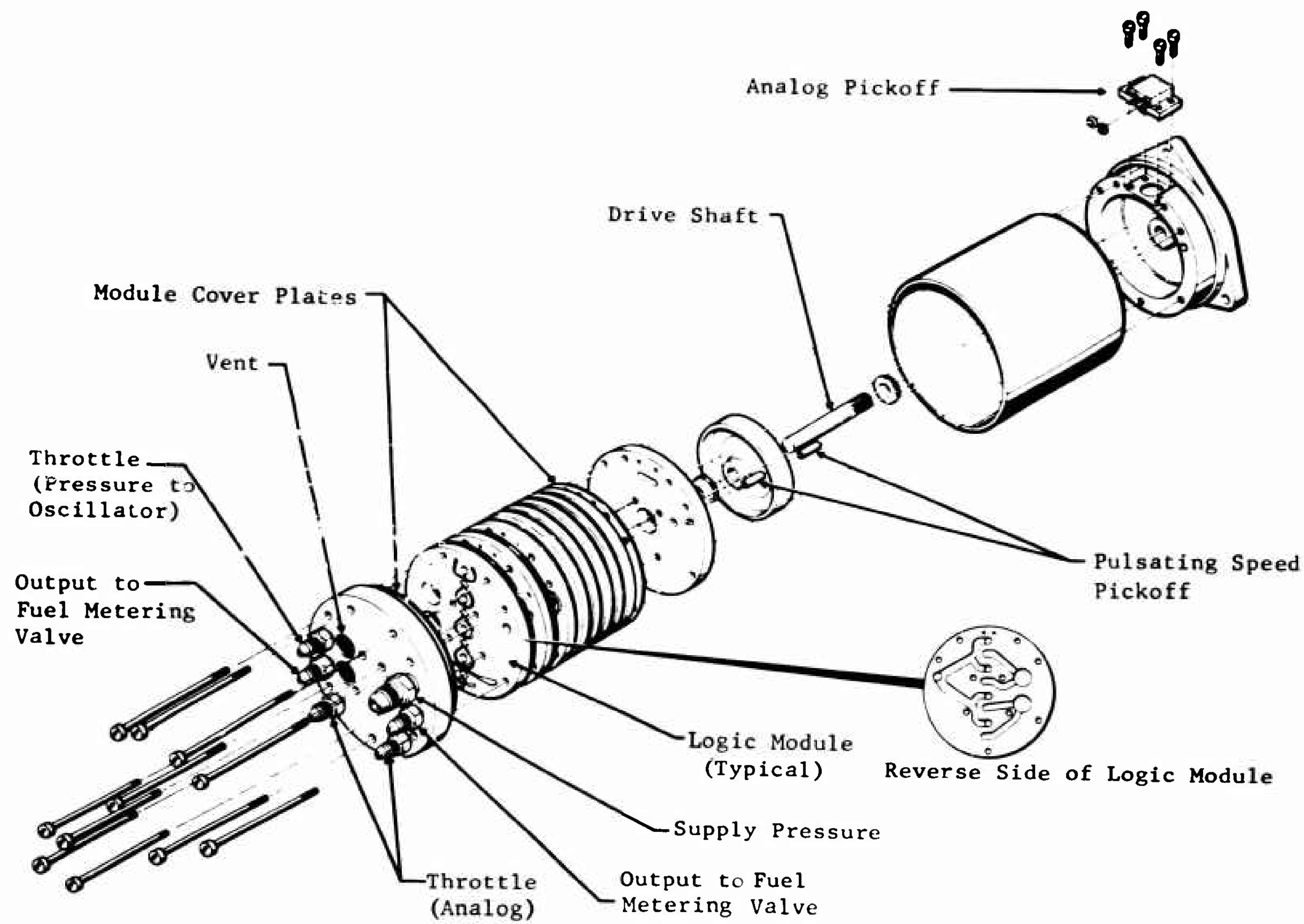


Figure 75. External Packaging Concept Using Modular Construction.

SIZE AND WEIGHT

Size and weight of a flightworthy fuel control will be influenced by the following factors:

1. Engine Application - environmental and acceleration requirements; use of variable compressor geometry and nozzle area control

2. Temperature Sensing and Display Requirements - temperature value; number of temperature sensors; number of display parameters

3. Future Sensor and Logic Miniaturization Developments - results of current development programs

The foregoing discussion of packaging approaches is intended to assist, in a general way, the earliest visualization of how to integrate the fuel control and engine. To achieve the optimum arrangement capable of meeting a specific requirement will require a team effort involving both the control and engine manufacturer in the layout phase.

POWER SUPPLY

REQUIREMENTS

Reliable operation of the fluid engine control system depends on finding a satisfactory power source for the fluid supply. Characteristics of the fluid power supply which are of principal interest include available pressure, pressure regulation, flow capability, temperature, and contamination. Satisfactory performance will generally have to be achieved in the presence of contaminated bleed air at compressor discharge pressure.

Control system requirements call for a steady flow in the range of 10 to 15 cubic feet per minute for sea level operation, at a pressure of 5 psig ±5 percent. Any transient disturbance in regulated outlet pressure due to a step change in pressure input shall return to steady-state value with a time constant of 50 milliseconds or less. Temperature regulation of the supply air is not believed to be necessary.

AIR-BLEED CAPACITY

Flow demand of the control system at sea level is about 0.0053 pound per second. This represents approximately 0.5 percent of the small engine compressor discharge at idle speed. Using power nozzles 10 mils wide for bistables, and 20 mils wide for proportionals, it is concluded that the flow required for a fluidic fuel control is not unreasonable for the small engine.

PRESSURES

Typically, the working head pressure available for the fluid power supply input is less than 5 psig for turbine engines idling on the ground at field altitudes of 3500 to 5000 feet. Less than 10 psig is usually available at all conditions of minimum flight airspeed or ground idle. Furthermore, part of the working head pressure will be dropped in the line filter and pressure regulator.

In some cases the fuel control system may need to be designed to perform satisfactorily at low rpm conditions at which time pressures at the fluid element supply ports will be approximately 2 to 3 psig. Also, with compressor discharge pressures varying over a range of 10 to 1, pressure regulation will be required.

Regulation - In addition to the factor of low pressure head available at idle, tests run to investigate the effect on fluid elements led to the choice of gage pressure regulation (i.e., regulation of ΔP across the fluid elements) rather than absolute pressure regulation of the power supply. Tests of fluid elements showed gross disturbances when supply pressure was held constant while outlet pressure was reduced to simulate

20,000- and 40,000-foot altitude. Superior performance was obtained when the pressure differential was held constant while the outlet pressure was reduced.

Maintaining the pressure head across the fluid system always at 5 psig means that mass flow through the constant-impedance fluid system will diminish with increasing altitude. Flow at sea level, 10,000 feet and 40,000 feet will be approximately 15, 10 and 6 SCFM, respectively. Fluid amplifiers are known to be sensitive to mass flow. If control flow and power flow are affected proportionately, the elements may tend toward self-compensation. Although this question is presently the subject of experimental investigation, it is of little importance for the operation of small engines at sea level or low altitude.

The bleed air is cooled considerably in passing through plumbing lines and a filter screen. Experience indicates that, in static operation at least, extraction air temperature will probably cool to 250°F or less before entering the regulator. In flight, the accessories in the forward half of the engine may be considered to be exposed to external temperatures commensurate with the ram air temperature rise. It is concluded that no difficult high temperature environment will likely be encountered in typical applications.

Since many of the fluid elements contain vent ports which are open to the atmosphere, internal static pressures in the fluid system are unlikely to fall much below ambient. So it does not appear that bleed air temperature and pressure conditions will fall to the dew point.

CONTAMINATION

There is a lack of long-term field operational or life test data to support design of filter protection for fluid elements. Present element designs have shown no contamination influences, but future miniaturized elements might be expected to be more susceptible. For this reason, contacts have been made with Honeywell and NASA MSFC personnel regarding field experience with contamination in the Honeywell air bearing gyro. Contamination problems with the air bearing gyro are several orders of magnitude more difficult than is expected for the fluid elements. This background does help, however, to point out areas where further investigation may be needed before final detailed specifications are written for the fluid power supply.

The air bearing gyro contains a spinning rotor on which air impinges from many small orifices. Rotor clearances are a few millionths of an inch. Particulate contamination is filtered by a composite stack of 0.45-micron millipore paper filters backed by wire mesh screen. These filters do not stop water or hydrocarbon vapors.

Operating life of the gyros was found to be limited by deposited, solid hydrocarbon tracks on the rotors. Hydrocarbon vapor in the nitrogen supply appeared to condense out on the rotor when dry nitrogen was throttled through a single orifice and impinged on the rotor. This problem has been alleviated by employing nitrogen gas from cryogenic sources and/or specifying dry nitrogen with less than 2 parts per million of hydrocarbon.

During self-powered engine operation, bleed air is expected to be relatively free of hydrocarbons except for possible lube oil leaks. Introduction of hydrocarbon vapors by impingement type starters would pose a possible source of significant contamination only if the auxiliary pneumatic power supply were furnished by a hot gas generator.

POWER SUPPLY SCHEMATIC

The schematic diagram of a possible fluid power supply is shown in Figure 76. This supply furnishes pneumatic power only to fluid control subcircuits. Any necessary bleed air required for pneumatic actuation is supplied from one of the remaining bleed-air ports.

Approximately 10 to 15 SCFM (at sea level) of bleed air is passed through an in-line, wire mesh filter to remove particles larger than about 40 microns in diameter. A pressure regulator referenced to atmospheric pressure maintains 5 ± 0.25 psig pressure head across the fluid control elements. Flow to the several fluid subcircuits is distributed by a supply manifold.

A relief valve provides automatic bypass should the filter become plugged with ice or contaminants. A bypass panel indicator is energized by the relief valve or by a transducer sensing pressure drop across the filter.

Field experience with contamination of compressor bleed air used to drive a conventional type pneumatic control has been gained by Allison Division on the T-63 engine. Subsequent engine tests showed that the degree of contamination varied by 100 to 1 depending on the location of the bleed-air port on the engine. Careful relocation of this port enabled Allison to eliminate the filter from the system.

PRESSURE REGULATOR

Honeywell Systems & Research Division has been investigating pure fluid methods of regulating pressure without resorting to moving parts. Results to date are summarized in Annex B. A practical method of achieving pure fluid, pneumatic pressure regulation is yet to be demonstrated. An RFQ was therefore submitted for a mechanical hot gas pressure regulator. Results indicate that a practical device meeting the RFQ specification can be developed.

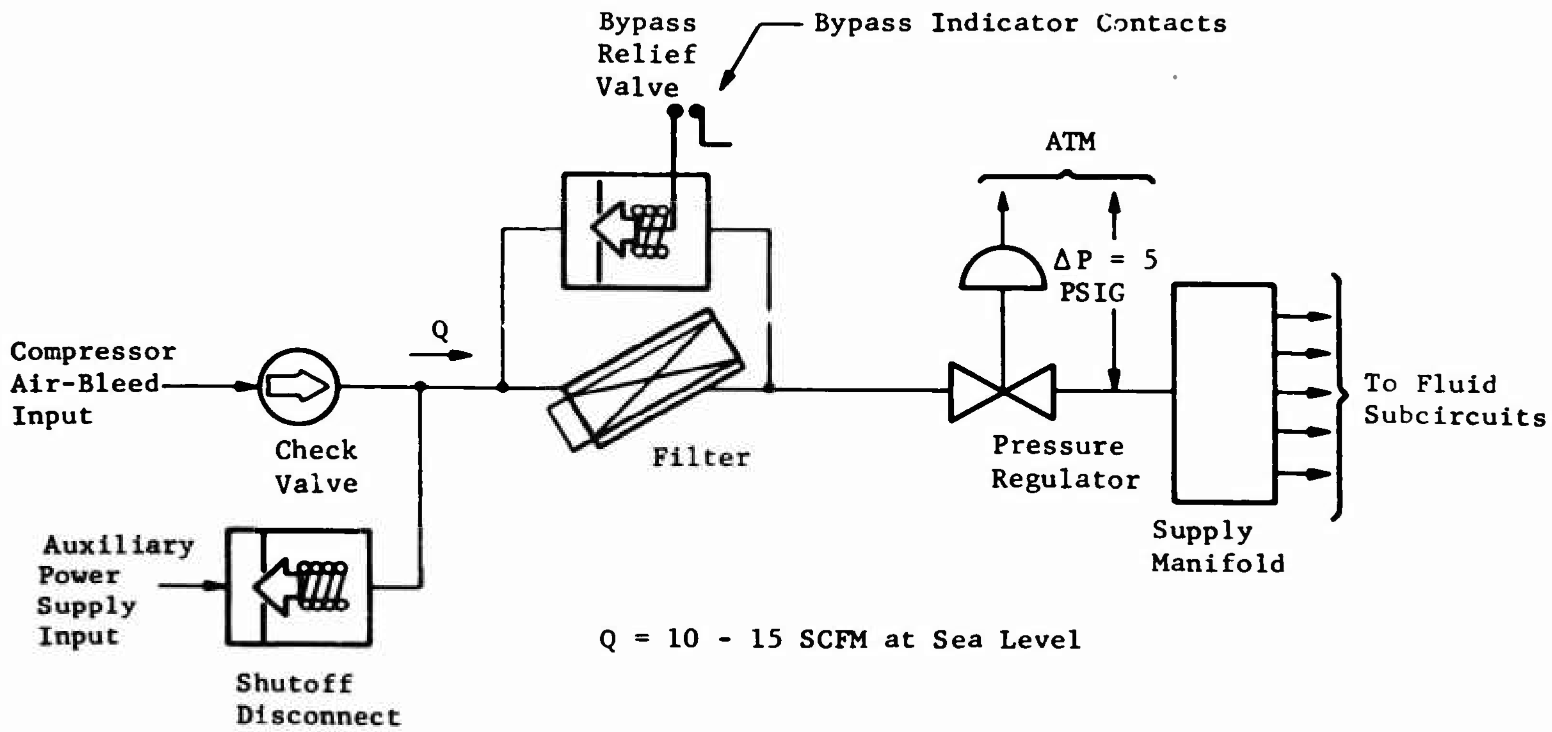
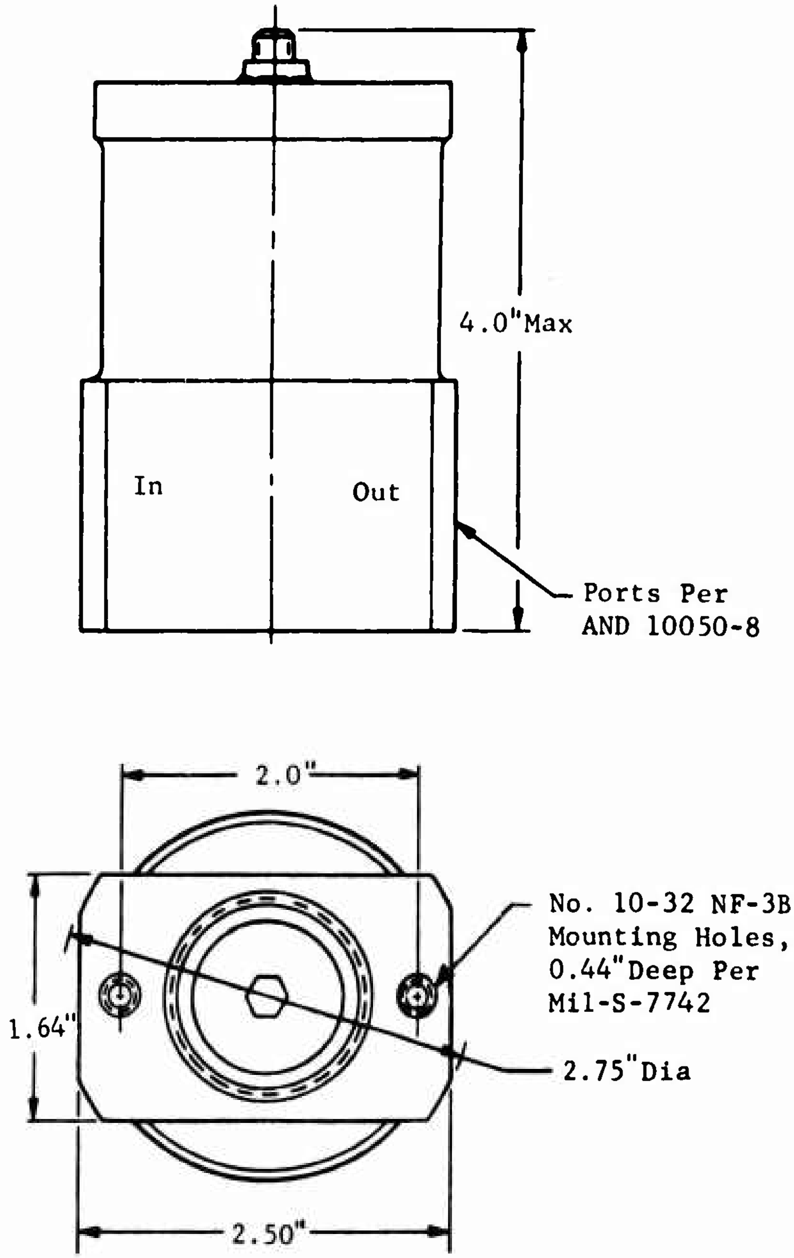


Figure 76. Schematic Diagram of Fluid Power Supply.

Figure 77 gives outline dimensions and an abbreviated specification for one solution as proposed by a qualified regulator manufacturer. This regulator represents a proposed new development employing a main valve that is a labyrinth seal piston working in a close-fitting main body. Employment of corrosion-resistant steels throughout and complete absence of moving seals enables the unit to be operated with bleed-air temperatures up to 850°F. The sensing piston is incorporated with the main valve and likewise forms a labyrinth seal with a larger bore in the body. This construction reduces the number of moving parts to one. Experience indicates that internal leakage or blow-by past this type of seal will not exceed 3 percent of the flow through the valve. This leakage is independent of inlet pressure and will have no material influence on regulator performance.

In order to avoid difficulty in a vibration environment, the pressure passage between the sensing area and the volume below the main valve is restricted. This restriction creates pneumatic damping and prevents excessive excursions of the main valve when the unit is excited at its natural or harmonic frequency.

The spring rate of the main reference spring is chosen so that the main valve will go from full closed to full open within the specified tolerance band of ± 0.25 psi. Flow area of the full-opened main valve exceeds the throughput area in the inlet and outlet fittings. Thus, due to no change in flow direction, pressure drop across the regulator is minimized when the inlet pressure drops below the specified outlet pressure.



1. Weight: 2.75 lb Estimated
2. Inlet Pressure: 150 psia to 5 psig
3. Proof Pressure: 225 psia
4. Burst Pressure: 300 psia
5. Outlet Pressure: 5 ± 0.25 psig
6. Fluid-Compressor Bleed Air
7. Operating Temperature: 0°F to 850°F
8. Flow: 15 SCFM at Sea Level

Figure 77. Hot Gas Pressure Regulator.

DISPLAYS

INTRODUCTION

As part of the small engine control study, consideration was given to methods of displaying engine temperature and speed without recourse to gear-driven transducers. The results indicate that purely fluidic methods of display are possible without gear reduction drives. Fluidic displays will most likely find application where very high display accuracies are not needed and where the weight of the fluid power supply (pressure regulator, filter, etc.) can be charged to another fluidic system such as an engine or flight control.

Some applications will probably need display accuracies of the order of 1 percent of full scale. Hybrid transducers having fluidic sensors and electronic signal conditioning circuits are capable of meeting this type of accuracy requirement. Spacious panel area allocations and highly readable display methods are needed to realize the benefit of very accurate sensors and circuitry.

In multiengine installations or anywhere that the instrument panel is widely separated from the engine(s), the weight of interconnecting plumbing and the display time-lag must be considered. These factors tend to support the electronic and hybrid approaches. In the case of fluidic temperature or pressure ratio sensing, either the fluidic or hybrid display approach must be used. Here an all-electronic approach is not applicable because of the fluidic nature of the signal sources.

Neither the electronic nor hybrid approach is applicable where temperature or radiation environments are incompatible. Fluidic displays are uniquely suited to severely hostile environments.

DISPLAY PARAMETERS

Selection of the parameters to be displayed will depend upon what information is needed to fulfill the basic mission and to prolong engine life. Some of the factors involved in this selection include:

1. Normal Mode - displays needed to monitor normal operation and to judge effects of aging or component failure

2. Manual Mode - minimal displays needed for pilot to control the engine manually without aid of automatic surge protection and temperature limiting

3. Environment and Mission - e.g., combination of salt-laden engines and short field takeoffs would require monitoring pressure ratio and engine speed or checking for overtemperature at a given engine speed

4. Sensor Integration - relative ease of integrating specific display transducers with the selected type of engine and fuel control

READOUT ERROR

There are at least three factors tending to limit the number and type of instruments devoted to display of engine parameters:

1. Available panel space
2. Limitation on number of simultaneous information bits usable by pilot
3. Cost and weight economy

In some present-day applications, the full accuracy potential of the engine transducer is not utilized in the display. In one J85 application, for example, a 3-inch-diameter indicator incorporates all of the engine displays including engine speed, temperature, and two flags. The speed and temperature scales are approximately 1.5 inches long and can only be read to the nearest 10-percent rpm; i.e., ± 5 percent. Since in this case the ± 1 percent MIL-Spec tachometer is compromised by limitations in available panel space, a less accurate, more economical display is justified.

ACCURACY ESTIMATION

In experimental work at Harry Diamond Laboratories,* proportional fluid amplifiers and passive fluid components were placed in simple circuits to perform operations of integration and summing. The experimental results varied from 2 to 17 percent below the theoretical predictions due to lowering of amplifier pressure gain caused by the relatively high output impedance of the jet deflection type proportional amplifier. The authors report that, by paying careful attention to loading effects, it may be possible to perform computation within ± 5 percent with present-day components. Passive components (fluid resistances and capacitance tanks) are required that are very close to linear over a wide range.

When sensor, indicator, and readout errors at environmental extremes are added to the above computational errors, it appears that pure fluid analog display systems may be expected to yield total errors of ± 5 to 10 percent of full scale. Future developments will probably bring the error down to about ± 5 percent. However, sea level static applications may usefully employ fluidic displays with perhaps half of the above error.

For applications requiring accuracies of about 1 percent of full scale, hybrid fluidic/electronic displays are recommended.

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*G.L. Raffman and S. Katz, "FLUID AMPLIFICATION - Experimental Fluidic Analog Computation," Harry Diamond Laboratories, Washington, D.C., TR-1292, 10 June 1965.

HYBRID DISPLAYS

Pulse Duration Modulation

Approach -- If the primary constraint in selecting the recommended method for indication was accuracy, electronic circuits capable of digital measurement of frequency or pulsewidth signals would be given first consideration. However, a fast-responding analog output is believed to fulfill best the information requirements for manual throttle actuation. The digital indicator has inherently higher accuracy than the analog type, but the information is in a less usable form for pilot proportional control functions. Digital phase measuring displays have satisfactory counting speed capability, but lack good trend information and also necessitate a compromise between repetition rate and readability.

The display circuitry described in the following paragraphs incorporates an analog type of display presentation with the inherent accuracy of pulse duration modulation (PDM) signal conditioning circuitry. The proposed PDM display circuitry has an estimated reliability one order of magnitude better than equivalent analog circuitry. Still, a compromise in reliability is involved as compared with pure fluid approaches. For applications where accuracy is not a stringent requirement and where the display location is close to the engine, a pure fluid display should be given consideration.

Pulse Duration Modulation -- In 1965, Honeywell built a breadboard model pulse rebalance servo for use in a capacitance type fluid measurement application. This device employs saturated digital circuits which are not affected by supply voltage variations from three to eight volts. As applied to pulse duration modulation for speed or temperature display, the circuit block diagram would take the form of Figure 78. The rebalance principle differs from the conventional servo in that the error signal is a pulse width as opposed to a voltage amplitude.

The indicator could take at least two forms - either a digital stepper motor or a torque motor indicator of the servometric type (being built by Honeywell for Lunar Excursion Module). As applied to a servometric indicator, the PDM circuit compares the width of two pulses applied to the opposite ends of the torquer (Figure 79). The residue pulses resulting from a difference in pulse widths are integrated by the torquer to provide an error signal which drives the pointer toward null. Width of the reference pulse is controlled by varying the resistance of an RC network in a pulse shaper. The pulse shaper function is similar to that of a monostable timer. Mechanical feedback of the pointer position adjusts the resistance to reduce the error signal.

Fluid sensors used to measure shaft speed or turbine inlet temperature provide input signal frequencies proportional to the measured parameter. The resulting fluid frequency signal is applied to a fluid-to-electrical transducer and shaped in a differentiator (if required). Pulse input rate is related to shaft rpm. As shown in Figure 80 , the period between pulses controls the dwell time of the flip-flop.

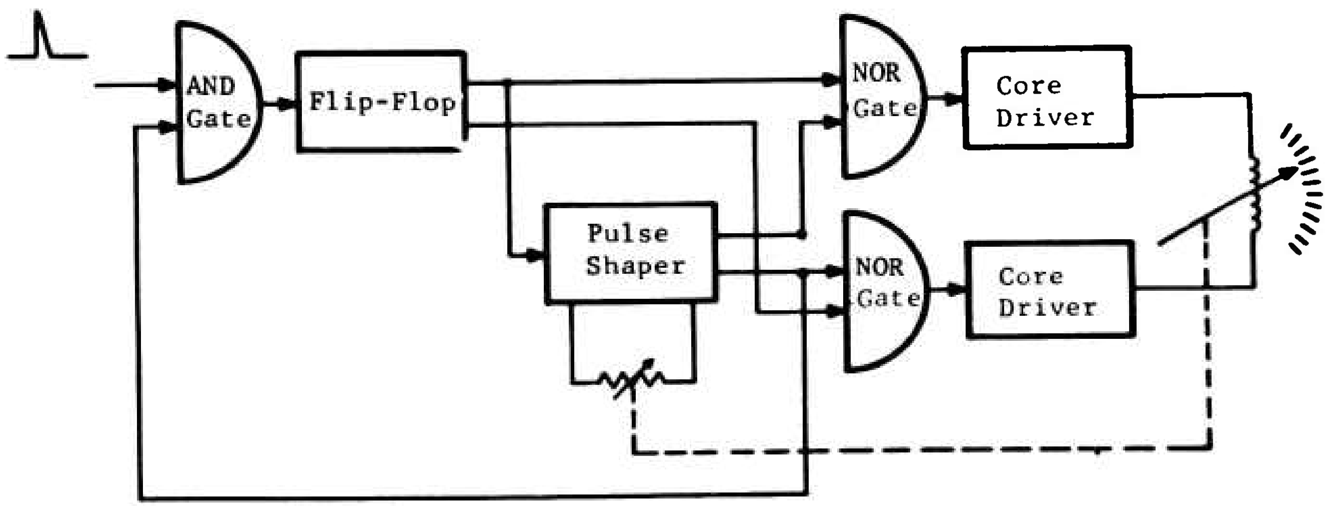

Figure 78. Speed or Temperature Display.

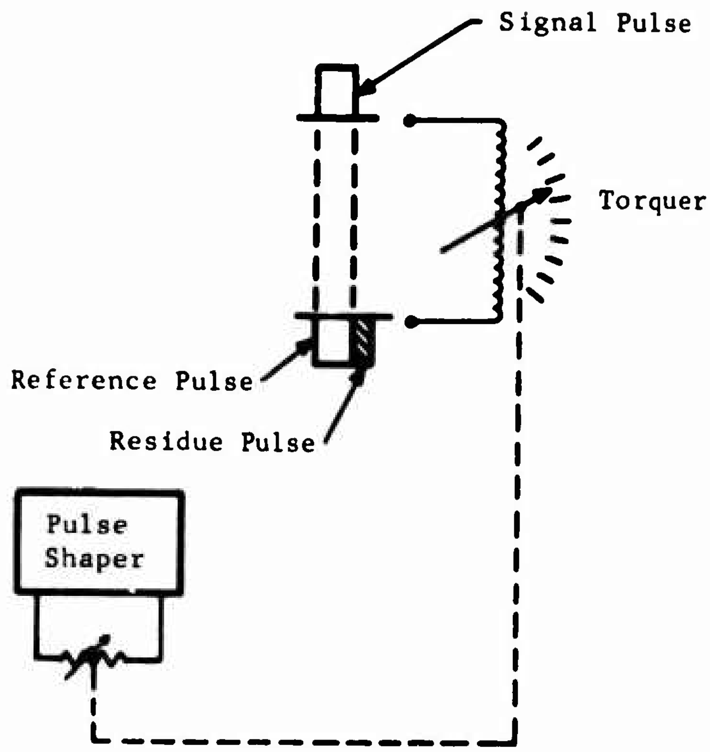

Figure 79. Integration of PDM Pulses for Analog Display.

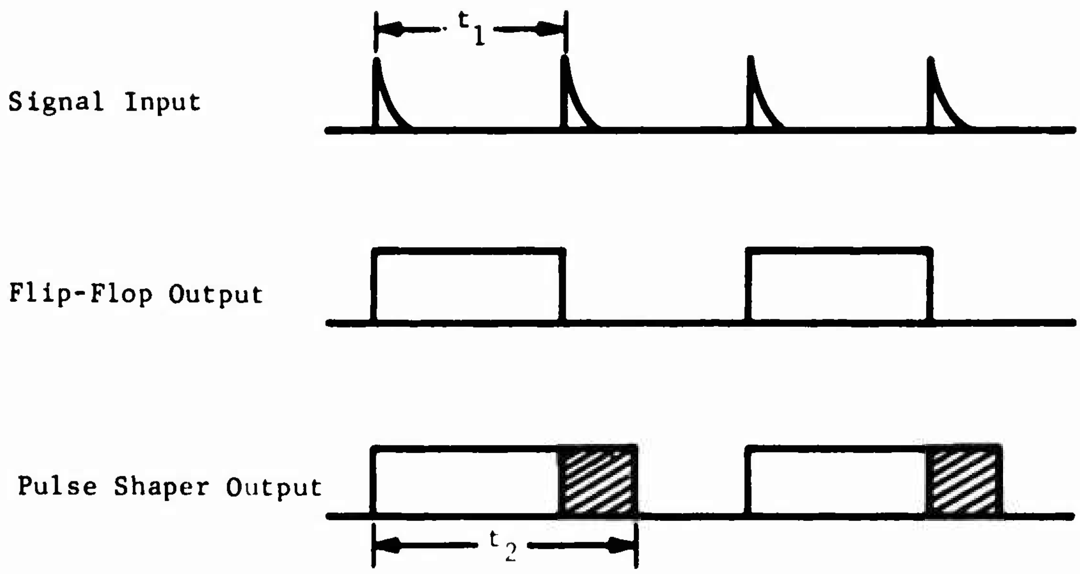

Figure 80. Signal Shaping Diagram.

Referring to Figures 78 and 80, the leading edge of the flip-flop signal turns on the pulse shaper; the pulse shaper turns itself off after a time (t_2) controlled by the position of the variable resistance. Feedback of the negative-going pulse from the pulse shaper to the AND gate (see Figure 78) blocks the input signal until the timed output from the pulse shaper (t_2) has passed. Signal dwell period from the output of the flip-flop is summed with the reference timer dwell in two NOR gates, the resultant pulse to the torquer having both duration and polarity intelligence. Core driver elements provide power gain for the resultant pulses driving the torquer.

Advantages

1. Rapid, analog, servo indication - no bouncing needle

2. Accurate, saturated digital circuits unaffected by supply voltage variations

3. Low power - less than 1/4 watt compared to 1 watt for equivalent frequency-to-digital circuit

4. Reliable, small package. With integrated circuits on four monolithic chips, reliability is 10 to 20 times that of analog equivalent

5. Could operate on a time-sharing basis with other signal inputs

6. For remote installations, electrical interconnecting cabling is lighter in weight and cheaper to install than tubing

Disadvantages

1. Requires very stable pulse shaper unaffected by temperature

2. For torquer method, a signal frequency of 1,000 cps minimum is required for good sensitivity

3. Circuit elements are limited to ultimate maximum temperature of 100°C

4. Compromises the simplicity and low cost potential of a pure fluid approach in favor of higher accuracy and lower weight

It is believed that a passive ramp generator such as that used in the Honeywell radar altimeter will provide the needed timer stability.

A nonoptimized breadboard has typically produced linearities ranging from 0.5 to 1.0 percent of full scale and sensitivities of 1.2 to 2.8 percent of full scale, depending on frequency. Since sensitivity is related to the average power developed across the torquer, an increase in frequency or supply power causes an increase in sensitivity. Frequency can be

increased by multiplying the pulses per revolution produced by the speed sensor. Supply power can also be increased, but at the expense of increasing the power dissipated throughout the logic circuit.

Fluidic/Electronic Frequency Converter

Approach -- Availability of pulsating or acoustic fluid signal outputs from speed, temperature and pressure ratio sensors led to a Honeywell-funded investigation of a microphone type transducer. The transducer had to be capable of the functions normally served by a quartz piezoelectric transducer and charge amplifier. Compared to the piezoelectric approach, however, it was desired that the new transducer be more economical, operate at higher temperatures and provide a direct readout. A breadboard model of such a circuit has been developed and is pictured in Figure 81.

Transducer Assembly -- An assembly breakdown of the transducer is illustrated in Figure 82. An acoustic or pulsating fluid signal displaces the stiff metallic diaphragm between two fixed plates of a differential capacitor.

Circuit Operation -- A block diagram of the fluidic/electronic frequency converter is shown in Figure 83. The differential capacitor is excited in push-pull fashion by a high frequency oscillator in the form of a three-wire capacitance bridge. The sensor is so designed that under quiescent conditions (no modulation), a slight, off-null condition is obtained at the bridge output. A carrier signal of low amplitude is present at this point. As the diaphragm is modulated by acoustical signals the null is changed and a modulated carrier signal is seen at the bridge output.

If a perfect bridge null were achieved, a typical output from a balanced modulator would be obtained as shown in Figure 84. This is difficult to achieve in such a device and in fact may not be desirable, as a simple diode rectifier modulator scheme would give frequency doubling. To prevent this possibility, the sensor is intentionally unbalanced to an extent greater than the maximum modulation capacitance change plus a safety factor for null drift. The resulting wave form is shown in Figure 85.

A differential circuit is employed to enhance the degree of amplitude modulation. If a simple 10 picofarad capacitor in series with an H.F. source were modulated by 0.1 picofarad change and the resulting signal were demodulated, a signal with 1-percent modulation would result. Assuming 20 db of carrier rejection, the differential capacitor technique would result in a signal of 100-percent modulation. The latter signal is more desirable and permits the use of a high gain preamplifier with a moderate dynamic range.

Referring again to Figure 83, the amplified signal is detected, filtered, amplified again and shaped, then used to trigger a one-shot multivibrator. The multivibrator provides electrical output pulses of fixed width and amplitude having a frequency the same as the repetition rate of the fluid input signal. Analog readout is provided by a D'Arsonval meter movement

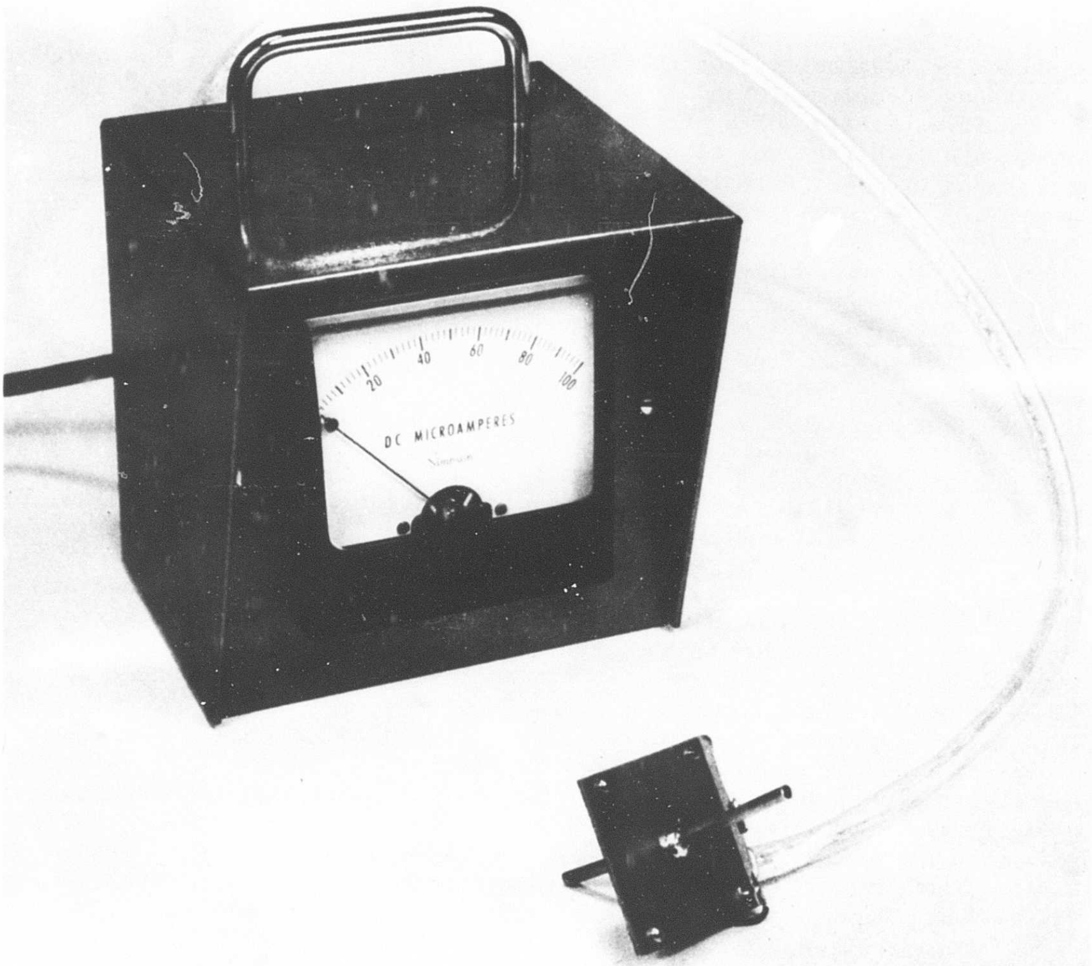

Figure 81. Photo of Breadboard Model Fluid-to-Electronic Frequency Converter.

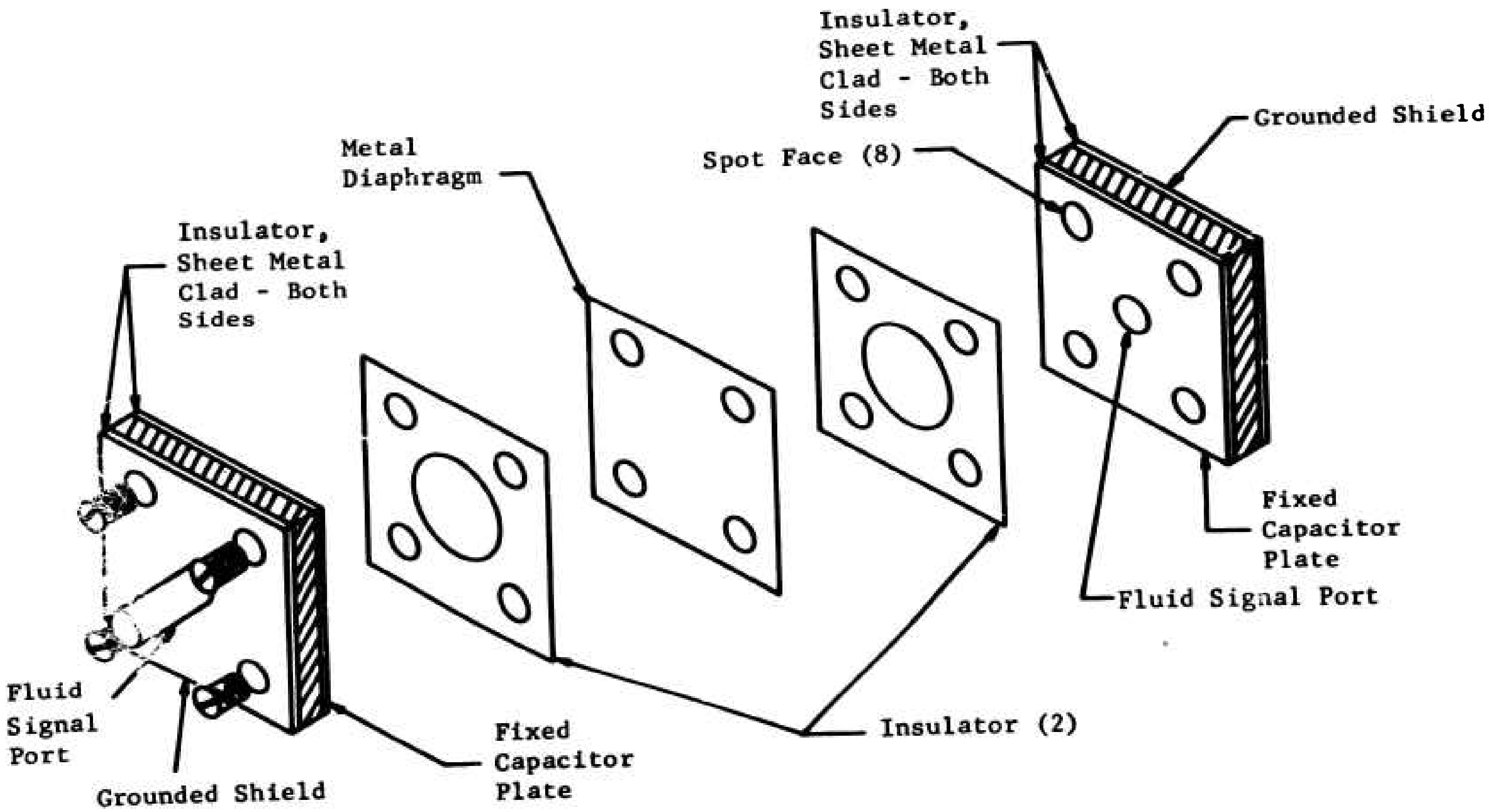

Figure 82. Transducer Assembly

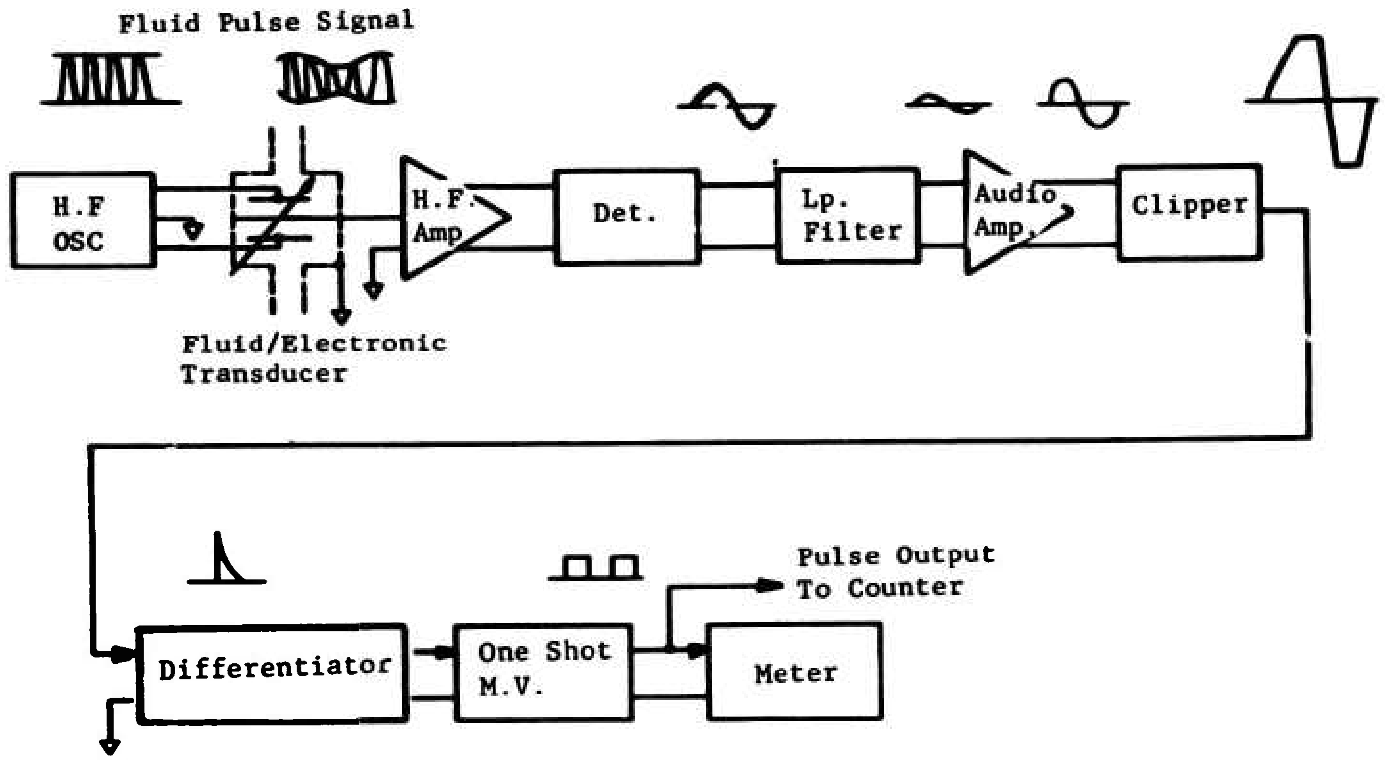

Figure 83. Fluidic/Electronic Frequency Converter.

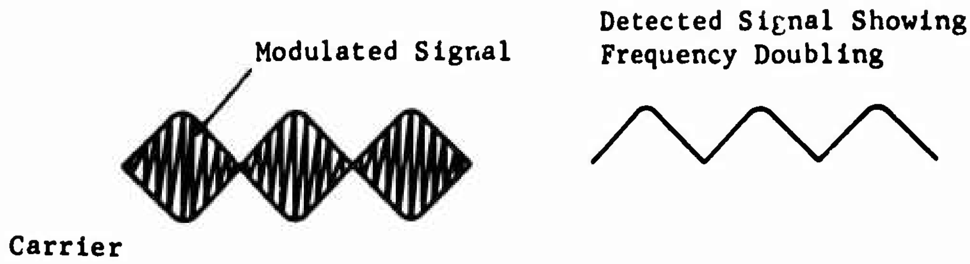

Figure 84. Balanced Modulator Output.

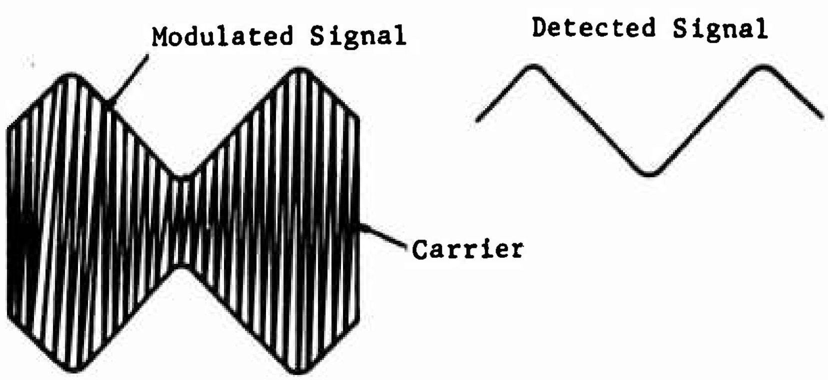

Figure 85. Modulator Output Intentionally Off Null.

which effectively integrates the current pulses. A pulse output is also available for counter use.

Summary Specification

Frequency range -- 50 cps to 10 kc
Minimum fluid signal level -- sensitive to voice power estimated at 60 db referenced to 10^{-16} watts/cm^2
Electrical Power Requirement
With external power supply -- 20 ma, 28v dc
With self-contained power supply -- 5w, 115v, 60 cps

Display Provisions
Suppressed zero for expanded meter scale
Provision for calibration with precision audio generator

ELECTRICAL SPEED TRANSDUCERS

Various types of electrical speed transducers are available for use in installations where environmental conditions permit. Since the characteristics of these units are well known, they will not be included in this report.

FLUIDIC DISPLAYS

Fluidic Analog Displays

Approach -- The present conception of an all pneumatic display employs fluidic sensing and signal conditioning with differential pressure gage indication. The signal conditioning circuit has application for reading out any signal source where fluid frequency is the sensor output variable. Immediate applications include temperature, speed (of the digital sensor type) and pressure ratio displays. The signal conditioner provides a d-c pressure proportional to the fluidic frequency output of the sensor and compensates for temperature sensitivity. The following circuit description refers specifically to analog readout of a temperature sensor but is generally applicable to pressure ratio and digital speed readouts as well.

Signal Conditioner -- The high frequency signal output of the temperature sensor is not usable with a normal fluid amplifier. Therefore, a frequency discriminating signal conditioner is used to convert the sonic frequency signal to a d-c fluid signal.

A schematic diagram of the temperature sensor and signal conditioner is shown in Figure 86. Output of the sensor is a fluid frequency proportional to $\sqrt{T_{abs}}$. The fluid frequency signal feeds into a resonator tube which provides a signal output whose amplitude is proportional to the input frequency. A fluid diode rectifies the a-c input from the resonator and,

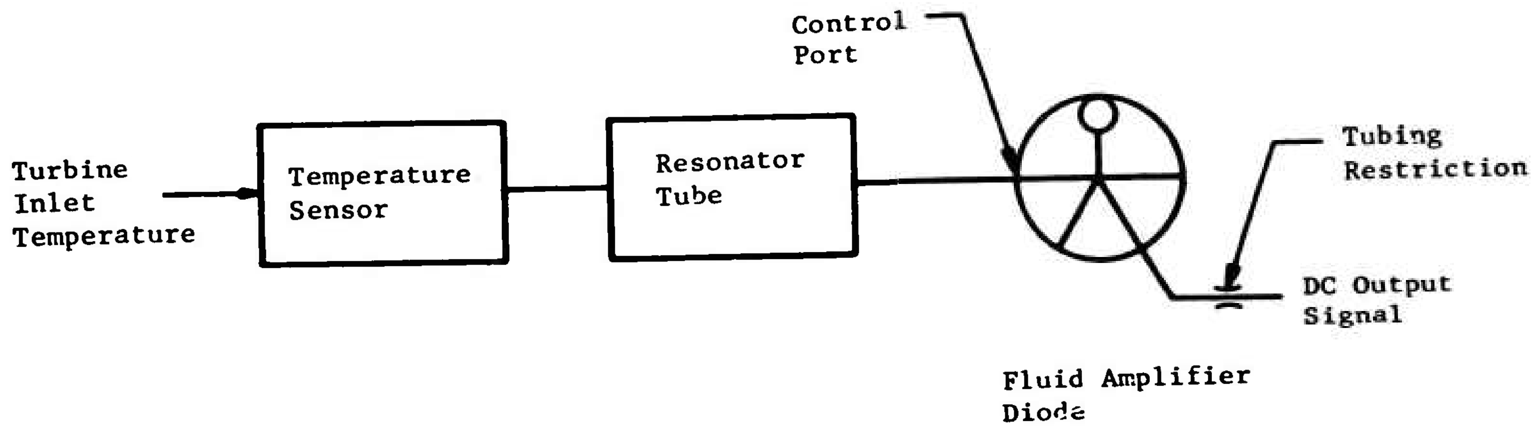

Figure 86. Temperature Sensor and Signal Conditioner.

together with the restriction afforded by a short length of tubing, provides a proportional d-c pressure output.

Resonator Tube -- The resonator is a tube of length equal to one-half the wavelength of the maximum expected frequency. Amplitude of the a-c signal increases as the signal frequency approaches the resonant frequency of the tube. Resonant frequency is determined by (1) length of the tube and (2) temperature of the gas.

Fluid Diode -- The fluid diode is constructed by careful design of exterior connections associated with a bistable fluid amplifier. The assembled diode stage operates to provide a rectified output proportional to the amplitude of the input signal. As shown in Figure 87, the amplifier is biased so that attachment occurs on the output leg nearest the input control port (see Figure 86). As the input signal swings negative, the wall attachment forces are increased and no output occurs. As the signal swings positive, the stream partially detaches from the near wall and a dynamic impact pressure is felt at the output port.

Figure 87 illustrates how the amplitude of the a-c control (input) signal, relative to the strength of bias, determines the degree to which the power stream will be deflected over to the output leg on the other side of the splitter. Amplitude of the input signal is never great enough to overcome completely the bias. Since the power stream is therefore never completely detached from the "near" wall, the power stream switches back to the "near" leg after the positive signal peak has passed.

A short length of tubing attached to the output port provides sufficient resistance to smooth out the rectified pulses. The d-c output pressure is thus a function of the a-c signal amplitude at the diode input. Figure 88 illustrates a typical characteristic curve for a resonator-fluid diode combination. Full-scale, differential pressure output of about 30 inches of water is available (reference figures). Output pressure can be doubled by using two discriminators in push-pull fashion.

Environmental Effects -- Limited environmental tests have been run on a display circuit consisting of a temperature sensor and basic, uncompensated signal conditioner. Output of the combined circuit was found to be sensitive to altitude changes as illustrated in Figure 89. These results are typical also of pulse-type speed sensing where similar signal conditioning circuitry is employed. Efforts are underway to compensate for or eliminate altitude sensitivity.

Temperature sensitivity of the signal conditioner is a characteristic which is used to distinct advantage for pressure ratio display, but is a nuisance for speed and turbine inlet temperature displays. The influence of temperature sensitivity on signal conditioner configuration is illustrated in Figure 90. In the case of the turbine inlet temperature and speed displays, signal outputs from the speed and temperature sensors are independent of ambient temperature. However, the resonator tube component of the signal

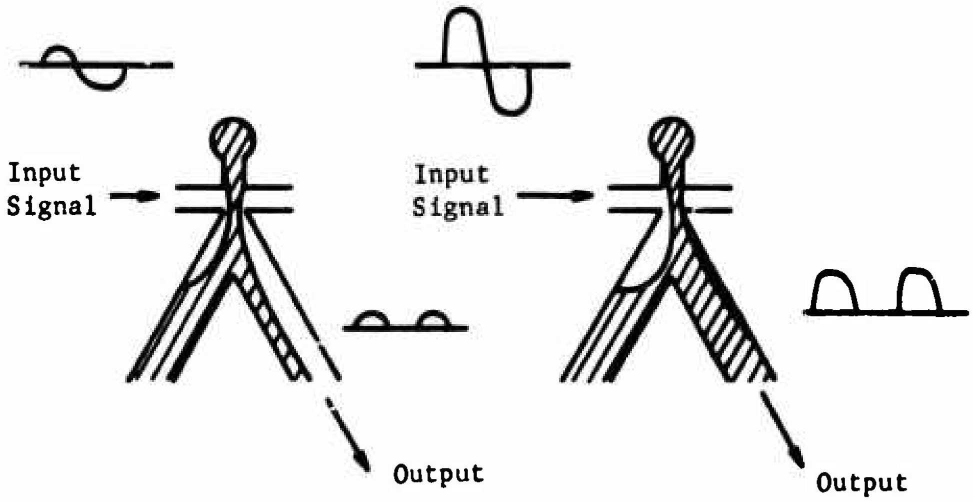

Figure 87. Output of Fluid Diode Related to Input Amplitude.

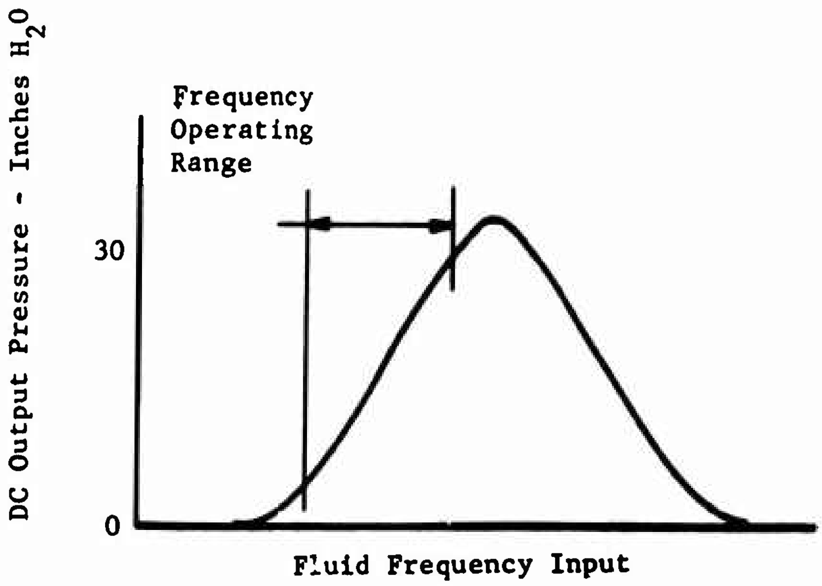

Figure 88. Output of Resonator/Fluid Diode.

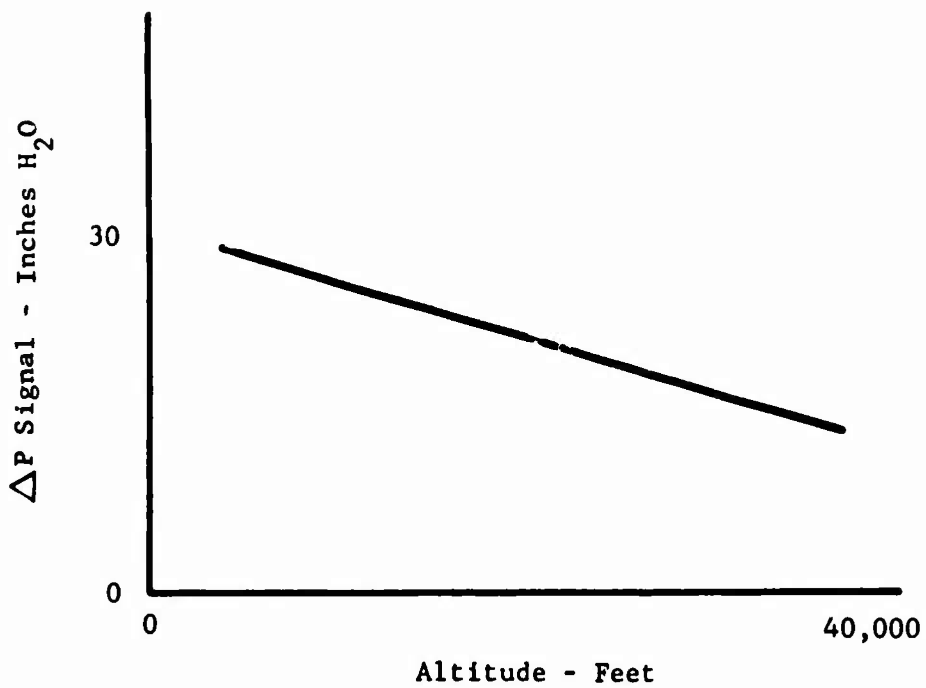

Figure 89. Altitude Sensitivity of Fluidic Temperature Display.

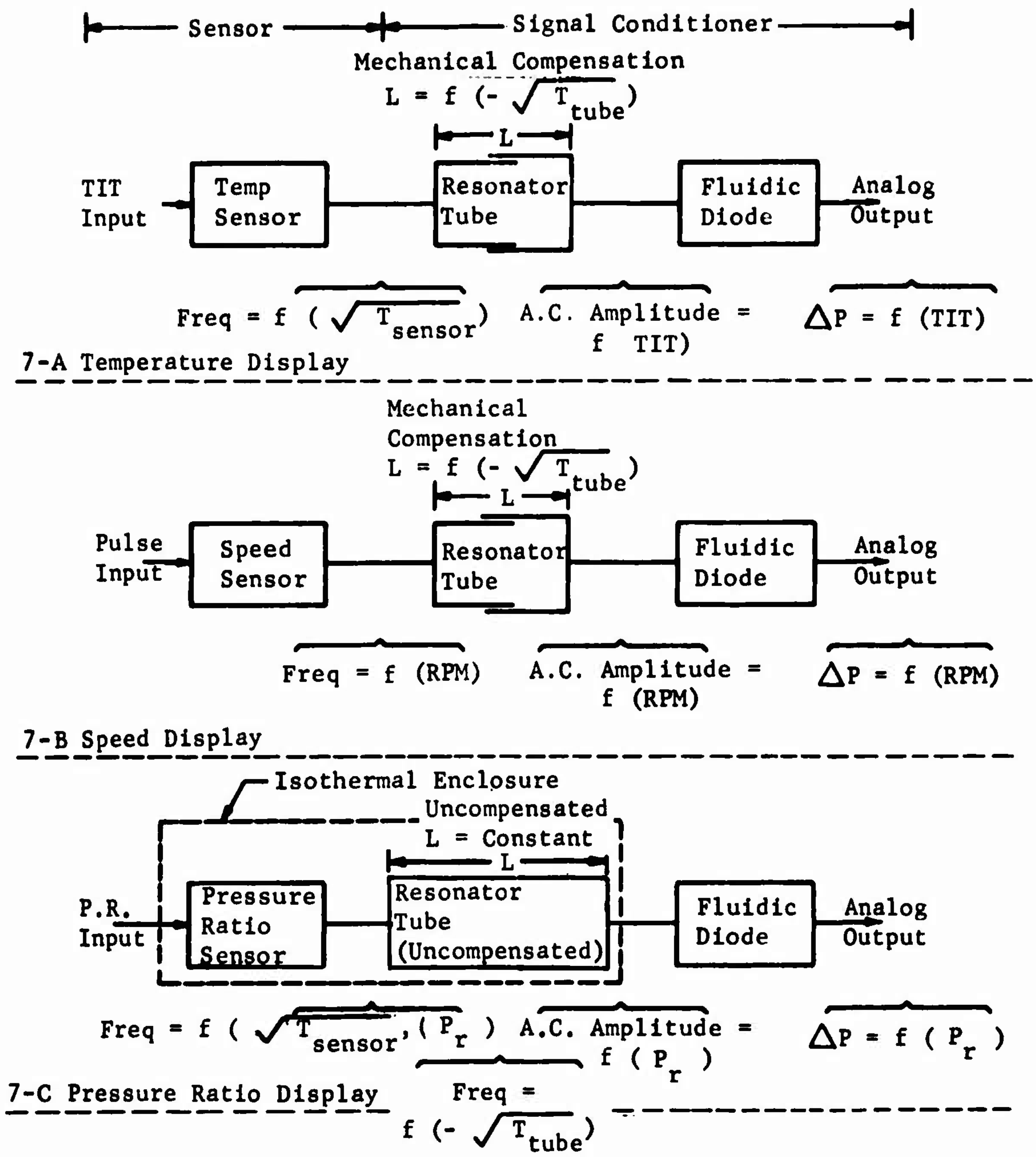

Figure 90. Influence of Signal Conditioner Temperature Sensitivity on Display Configuration.

conditioner is sensitive to ambient temperature. One approach being considered for compensating the resonator tube is to adjust its length as a function of $\sqrt{T_{abs}}$ where T is the absolute gas temperature in the tube.

Temperature compensation is an inherent feature of the fluidic pressure ratio display. It happens that the frequency output of both the pressure ratio sensor and the resonator tube changes as a function of the square root of the absolute gas temperature. These two effects are in opposition in the combined pressure ratio circuit (see Figure 90) so that the temperature sensitivity is cancelled. Since the gas temperature must be kept the same in both the transducer and the resonator, an isothermal enclosure is shown schematically in Figure 90. Temperature compensation and frequency detection are thus accomplished in a single circuit.

Figure 91 shows how the temperature compensation works in the pressure ratio display circuit. Suppose the resonator is set for a resonant frequency of fr_1 at room temperature and the pressure ratio output of the fluid diode is at (a). If the inlet temperature of the pressure ratio sensor changes such that the frequency increases, the output of the fluid diode will tend to increase to (b). However, the increased temperature will increase the resonant frequency of the resonator to fr_2. At this new resonant frequency, the output will be at (c), which is the same as the original state.

Fluidic Status Display

Honeywell has developed and is presently evaluating a fluidic/mechanical indicator. This device, dubbed the Honeywell fluidot indicator, can be used in pairs, one red and one green. The fluidot indicator provides a strinkingly effective display and affords a small, lightweight, pneumatic method of indicating flameout, status of sequence functions, or preselected levels of speed or temperature.

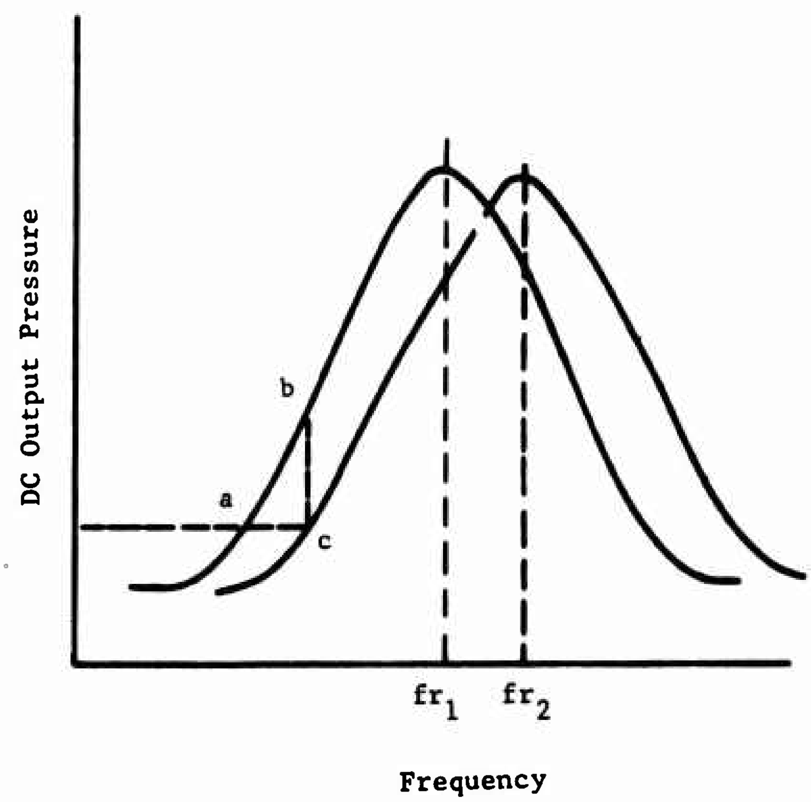

Figure 91. Temperature Compensation of P.R. Transducer by Resonator.

RECOMMENDATIONS FOR FUTURE WORK

The work reported here can be considered a Phase I pilot study, now completed. An appropriate follow-up to this study should be made by applying the hardware techniques being gained on the USAF Integrated Engine Control Contract to the particular problems of the small engine installation.

It is recommended that two fuel control and display systems be fabricated and their commonality aspects investigated; one including a very simplified fuel control, the other a more advanced fuel control. These systems should be demonstrated under sea level static conditions on the J85 engine, since the dynamics of this engine are very similar to those of the small engine, and the J85 is already included in the Honeywell engine test facility. A third advantage of this approach is one of economy. The engine map computation, bleed and nozzle area control schedules and engine computer simulation have already been done for the J85. The principal disadvantage of using the J85 for demonstration is that it does not include a free-coupled load turbine.

The following task statement briefly defines the suggested demonstration work as well as a design study to further the integration of the fuel control and start and relight sequence control with a paper engine to be specified by Curtiss-Wright.

STATEMENT OF WORK

Phase II - Integration of Fuel Control With Paper Engine

1. Compute engine map specifically applicable to a small engine.
2. Determine control system configuration to meet specification requirements supplied by Curtiss-Wright for
   - Engine mission
   - Throttle burst
   - Selection of fixed or variable control for compressor bleed and nozzle area
   - Schedule specification if variable area is used
   - Display variables and required accuracy
   - Environment
   - Starting power source and start and relight automation requirements
3. Conduct analysis and simulation as may be required by new engine map or area control schedules.
4. Conduct limited component feasibility study to supplement results of USAF integrated engine control contract. Includes investigation of frequency dividers (fluidic gear train) to reduce output frequency of high frequency signal sources.

5. Participate with Curtiss-Wright in laying out suggested locations for fluidic sensors and integrated circuit packages.

Phase III - Fabrication of Two Fuel Control Systems and Demonstration on J85

1. Determine fuel control and display system requirements for two applications as defined by Curtiss-Wright. Applications may differ as to engine mission, throttle burst requirement, use of variable bleed and nozzle area controls, and fluidic versus hybrid displays.

2. Design and fabricate demonstration models of a simplified fuel control, a start and relight sequence control, a more advanced fuel control, and two different display systems. Enhance commonality aspects of the two fuel controls, where possible.

3. Conduct sea level static demonstrations of the fabricated systems using the J85 engine.

ANNEX A

ENGINE DYNAMICS COMPUTATION

INTRODUCTION

This section gives a description of the method used in computing the engine dynamics. Conclusions and results were given earlier under the heading COMPUTED ENGINE DYNAMICS.

APPROACH

The acceleration time constant is desired for the gas generator portion only. Since there is no mechanical load connected to this shaft, and since the diaphragm or nozzle to the output turbine is of fixed area and running choked, the dynamics and the thermodynamics of the gas generator are essentially the same as those of a simple turbojet with fixed nozzle.

COMPUTATIONAL PROCEDURE

The following procedure was used to compute the dynamic characteristics of the engine:

1. The turbine flow line in Figure 92 can be shown to be a hyperbola for any engine. Since the flow must be zero at a pressure ratio of 1, it is only necessary to know the flow rate, burner temperature, and engine pressure ratio at one other point, the design point, to define this curve completely. The pressure ratio at the design point was estimated by assuming a 7.5-percent pressure loss in the burner.

   Gas flow rate to the turbine was estimated from the compressor air rate by subtracting the cooling air allowance and adding the fuel flow rate.

2. Several different burner temperatures may be assumed, and for each one, values of the flow parameter can be read from the flow line in Figure 92 at selected pressure ratios, and the corresponding flow rates can be calculated and plotted as a constant temperature line in Figure 93. The constant speed lines and the surge line of Figure 93 are taken from compressor data furnished by Curtiss-Wright. The engine performance map of Figure 93 is then essentially a grid of speed lines and temperature lines.

3. It is possible to determine the turbine pressure ratio for any given overall pressure ratio and flow rate by trial and error. However, it is easier and faster to plot this relationship and read the turbine pressure ratio directly. In general, this would

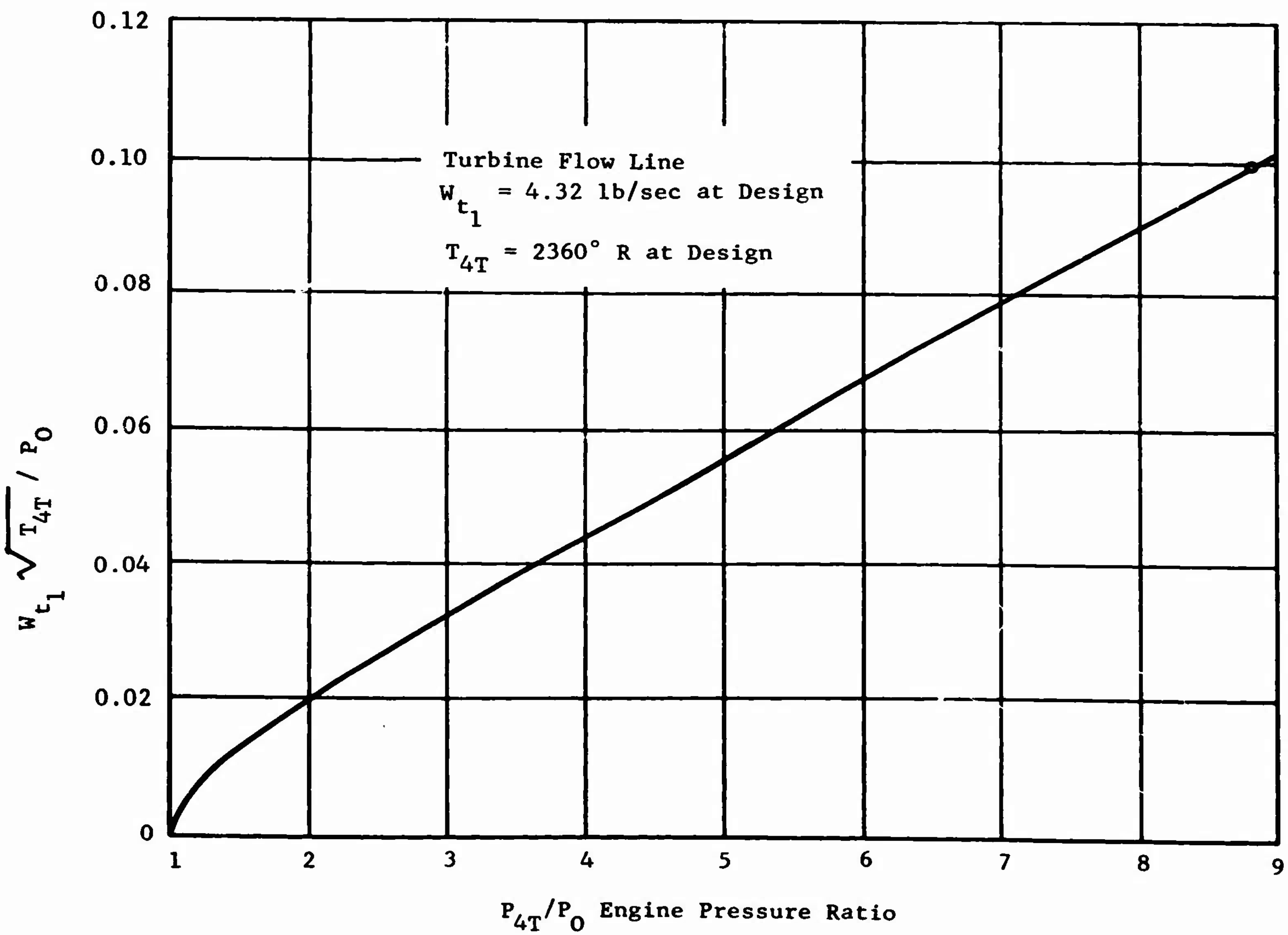

Figure 92. Turbine Flow Line - Flow Parameter Versus Pressure Ratio.

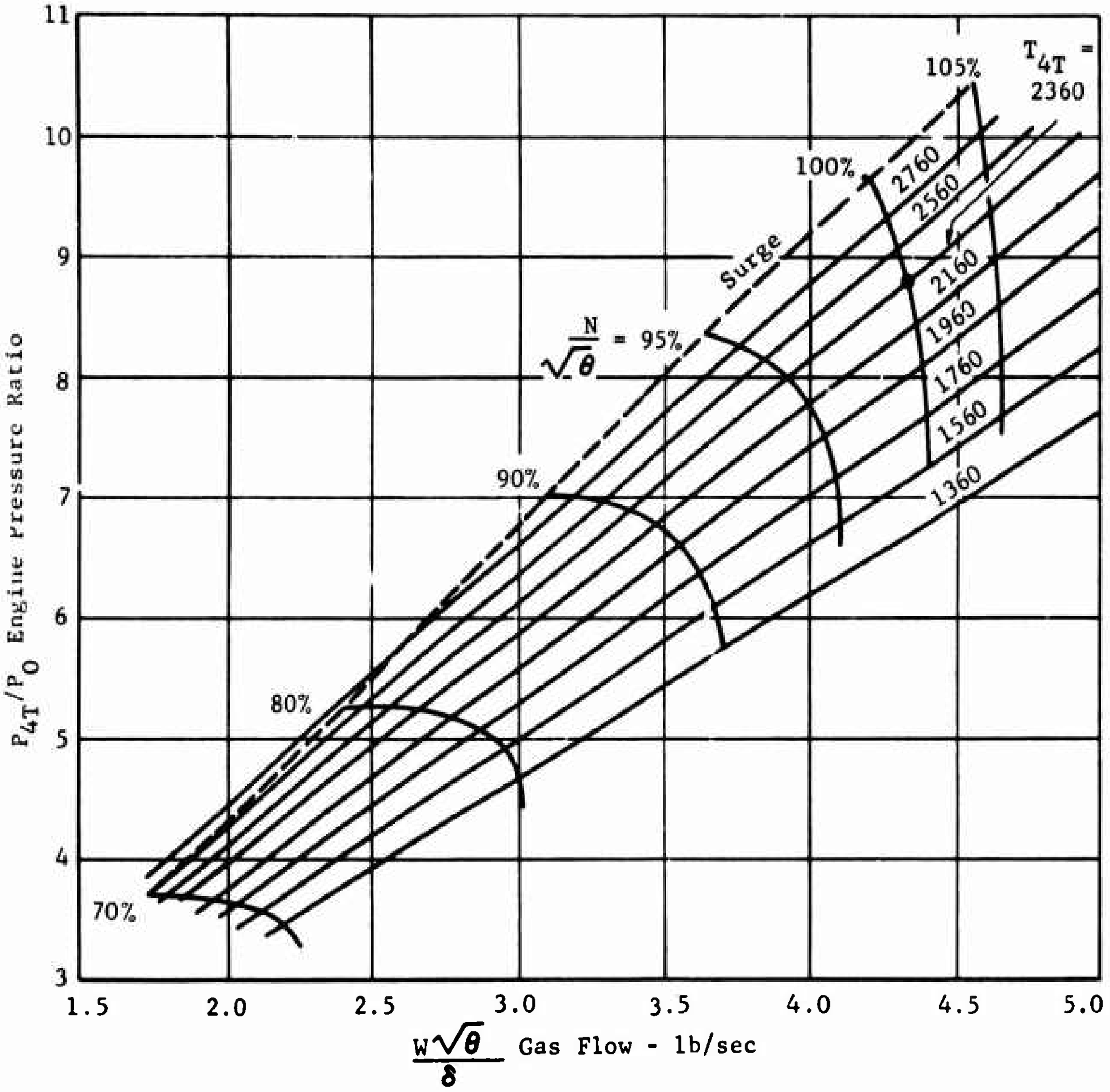

Figure 93. Engine Performance Map - Pressure Ratio Versus Gas Flow at Constant Temperatures.

be a family of curves for various overall pressures ratios, but Figure 94 shows only the choked nozzle portion of these curves, which is sufficient for this problem.

4. There is now enough information for calculation of the fuel flow rate, the compressor power, and the turbine power of the gas generator at any point on the chart of Figure 93. It will be sufficient to do this at the intersection points on the grid, thus avoiding interpolation errors. The difference between the compressor power and turbine power can be expressed as a net accelerating torque on the gas generator shaft. This torque will be zero at the steady-state operating point for each speed and will be positive at higher temperature and negative at lower temperature.

5. The results of the above calculations may be plotted in many ways for different purposes. For our immediate purpose, it is sufficient to plot torque versus fuel flow at constant speeds, as in Figure 95, and to cross plot these to get torque versus speed at constant fuel flows, as in Figure 96.

6. The slopes of the torque-speed curves in Figure 96 are a direct indication of the acceleration time constant of the engine, as follows:

$$\tau_e = \frac{-I}{\left.\frac{\partial Q}{\partial N}\right|_{W_f}}$$

Since the slope, or partial derivative, is negative, the time constant is positive and is in units of seconds if proper dimensions are used. The dimensions are as given in the List of Symbols.

The time constant of the engine may be determined from the above equation at any point in Figure 96 by measuring the appropriate slope. For our purpose, it is sufficient to do this only at points along the zero-torque axis, or steady-state conditions. This gives the time constant at any steady-state speed, as plotted in Figure 93.

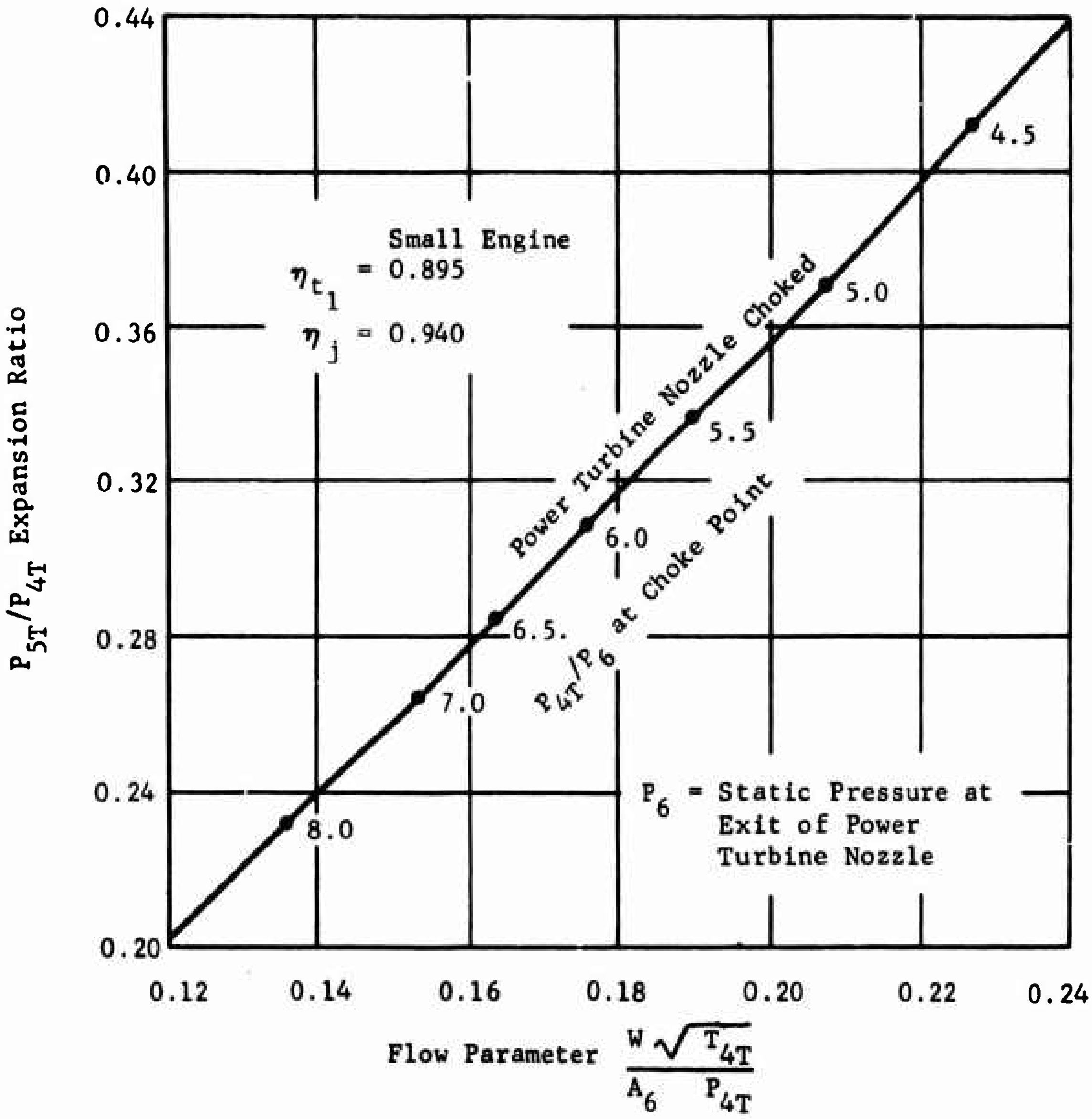

Figure 94. Expansion Ratio Versus Flow Parameter.

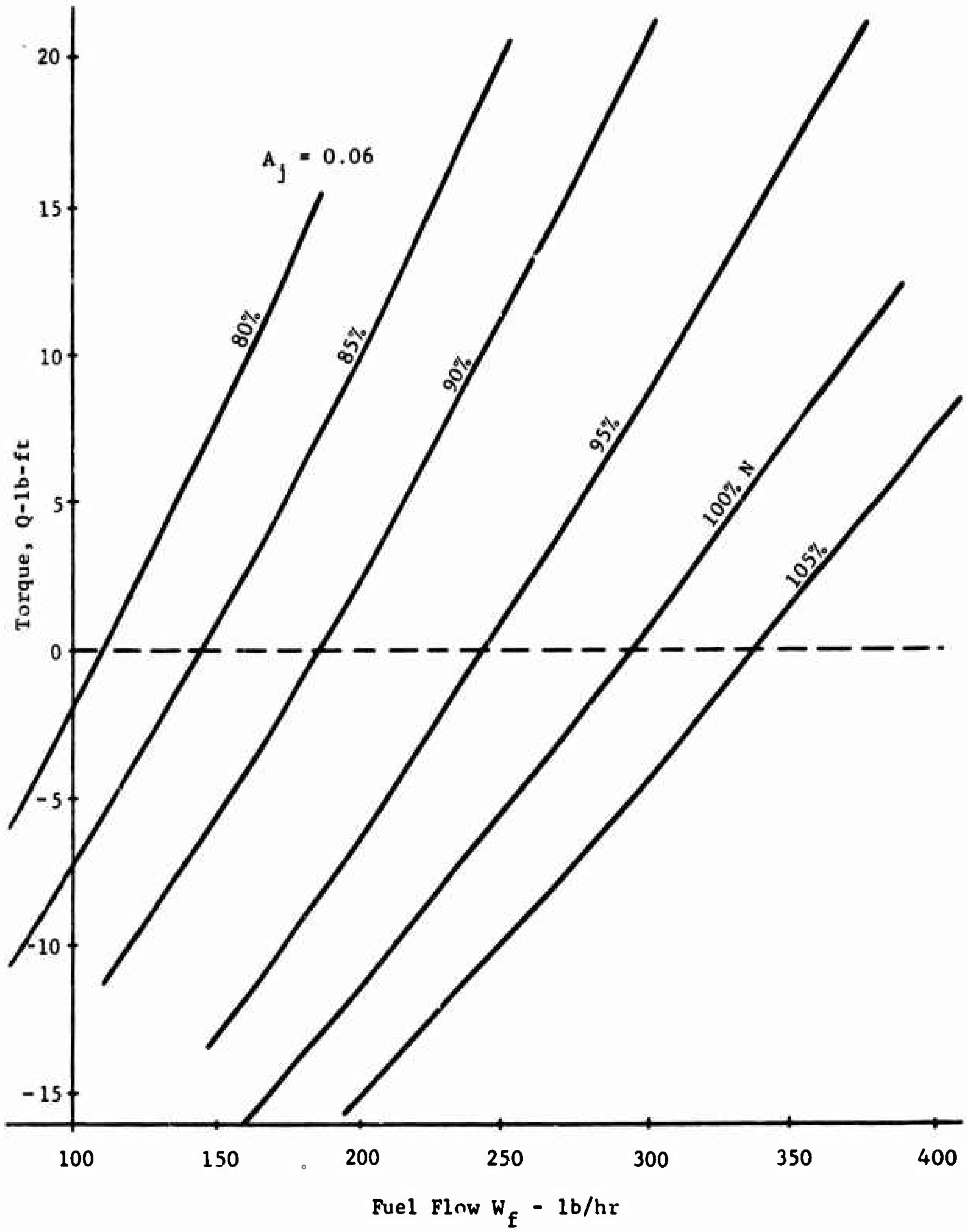

Figure 95. Torque Versus Fuel Flow at Constant Speeds.

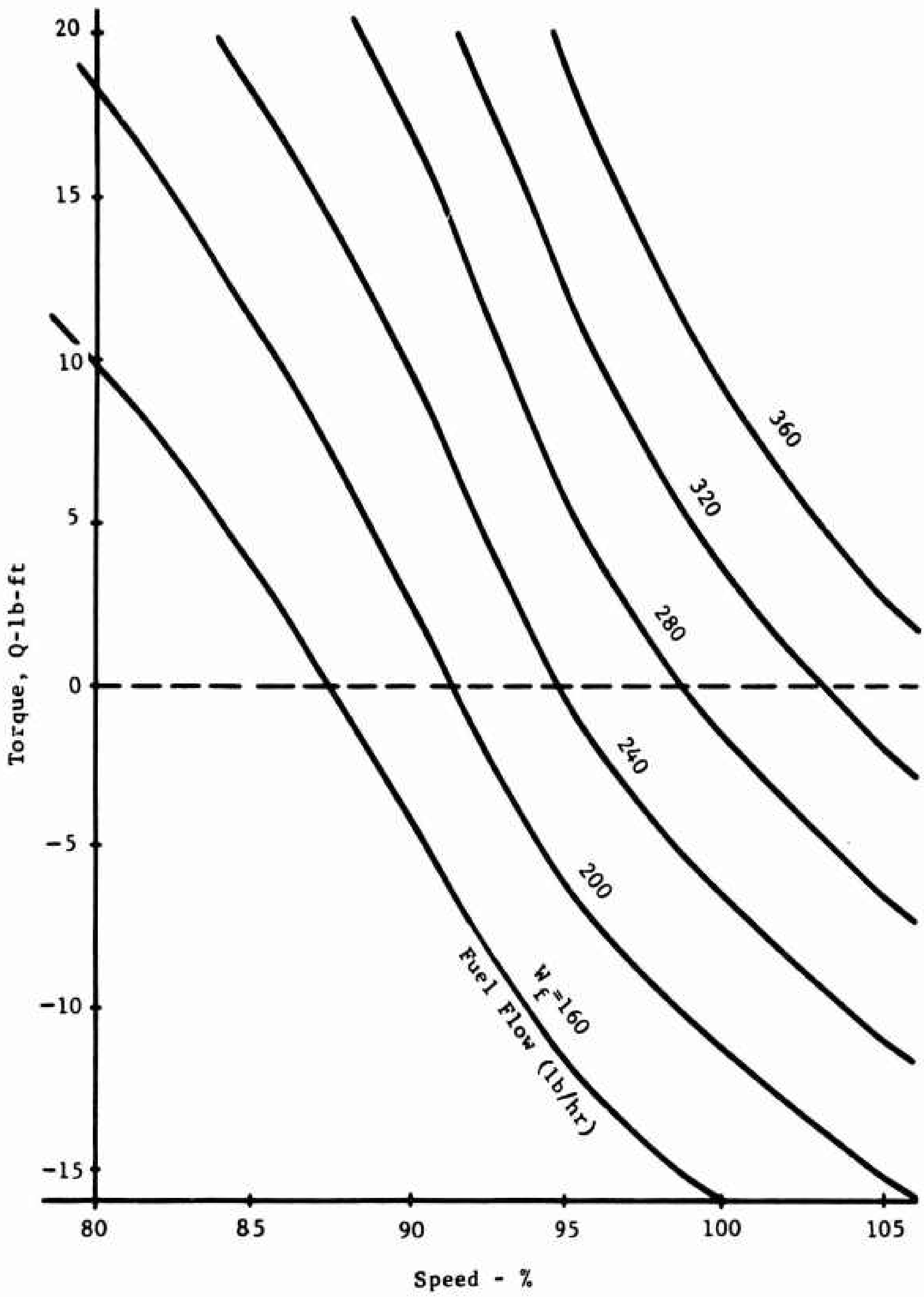

Figure 96. Torque Versus Speed at Constant Fuel Flows.

ANNEX B

FLUID AMPLIFIERS

Fluid elements are generally a configuration of flow channels and orifices through which the operating fluid flows. The interaction of flow streams, entrainment of adjoining fluids, the directing effect of walls, the resistance of orifices, and the capacitances of volumes produce effects which are the intelligence of the fluid elements and can be used or measured. These effects are: proportionate or bistable switching of flow from one channel to another; increase in rotational energy; or pulsing frequencies which are a measure of some environment (i.e., temperature or pressure). These effects are used in pure fluid systems or are transduced to an electrical or other signal for hybrid applications.

Since the intelligence is obtained from the fluid flow phenomena without the assistance of moving mechanical parts, fluid elements are simple, easy to construct and reliable, provided that we have a reliable power source and noncontaminated fluids. If air is used, the system is capable of operating at extreme temperatures, in nuclear radiation or in severe vibration and is only moderately influenced by high "g" forces or shock. Examples of typical fluid elements are represented in Figures 97 through 101.

BISTABLE AMPLIFIERS

The right-hand element in Figure 97 is a bistable amplifier with power ports, control ports and output ports indicated. When power is applied, output flow occurs in one output port or the other -- not in both simultaneously. A lower pressure (or flow) control differential will switch the flow. If the control signal is removed, flow will remain in the channel to which it was directed. Therefore, the operation is bistable. The memory characteristic makes it useful as a computer element in addition to its obvious use as a bistable power amplifier or flow switch.

The center and left-hand elements in Figure 97 show the Coanda effect which accounts for the bistable action. A jet issuing between two walls (as on the left) will entrain air from the areas between the walls and the jet due to viscous forces. This pumping effect reduces the pressure in these areas - more on one side than the other because perfect symmetry does not occur. The resulting differential pressure deflects the jet, reducing the inflow of new air to replace the entrained air. The net effect is unstable, and the jet is forced to one side where it appears to attach to the wall. The addition of control ports allows the phenomenon to be controlled and provides the now familiar bistable amplifier. Figure 98 shows several sizes of bistable amplifiers.

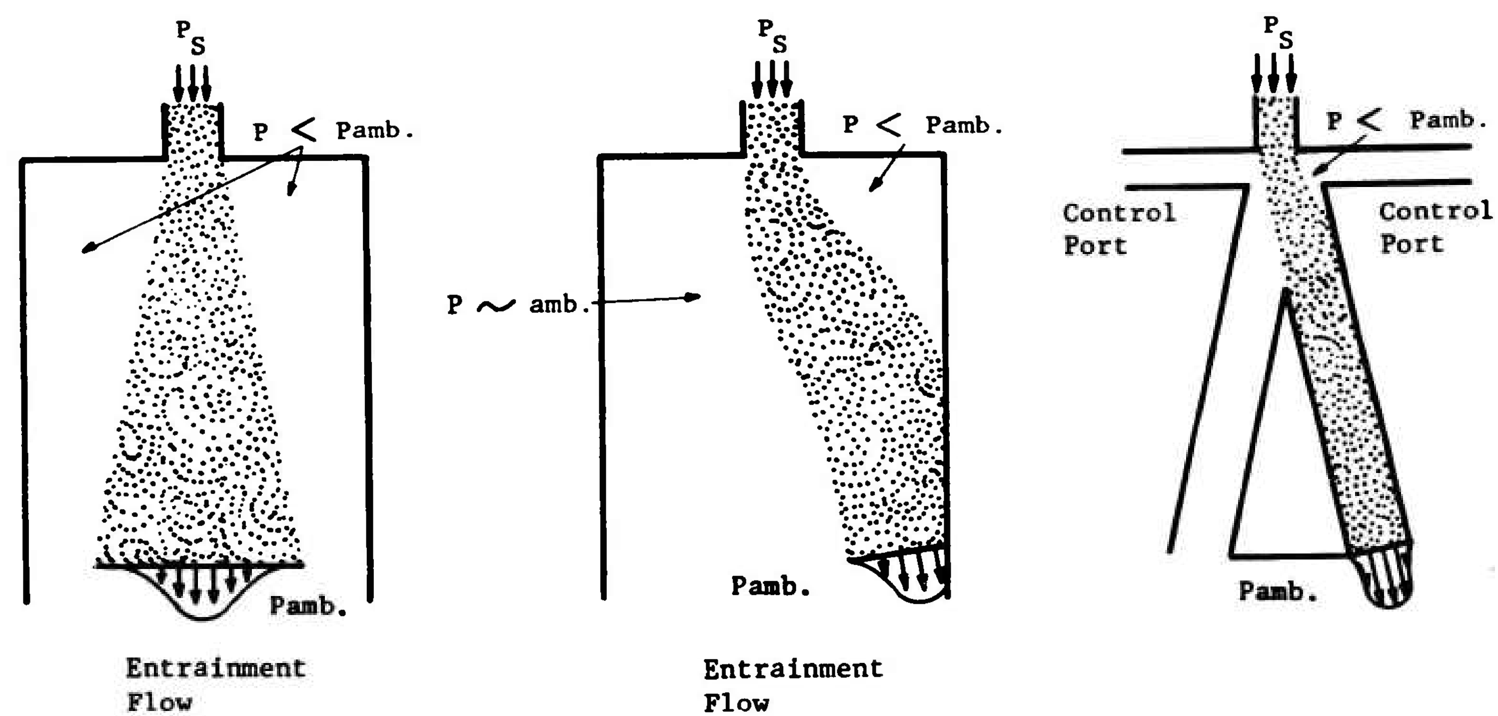

Figure 97. Bistable Amplifier Wall Attachment Phenomenon.

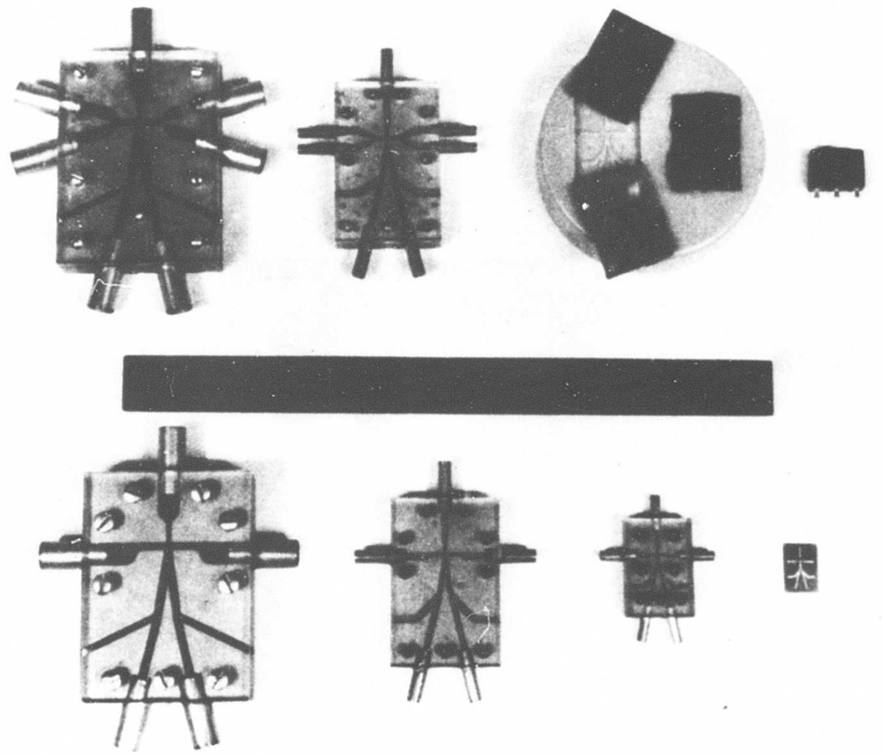

Figure 98. Bistable Fluid Amplifier.

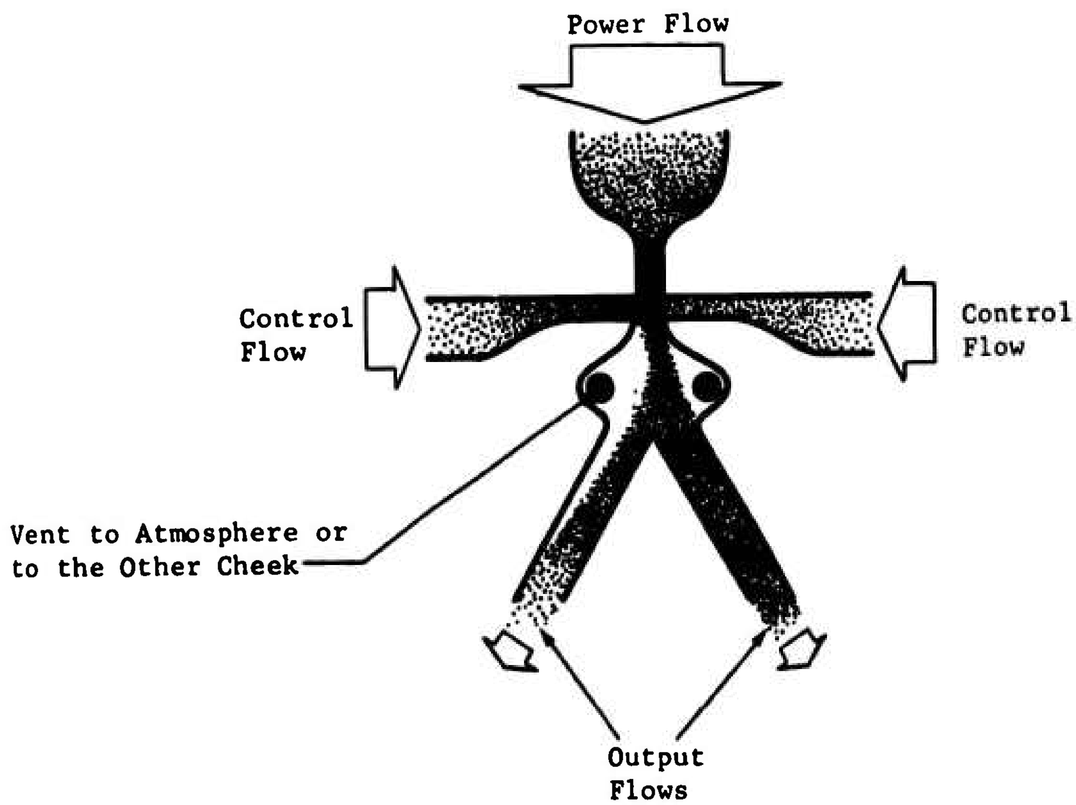

Figure 99. Proportional Fluid Amplifier.

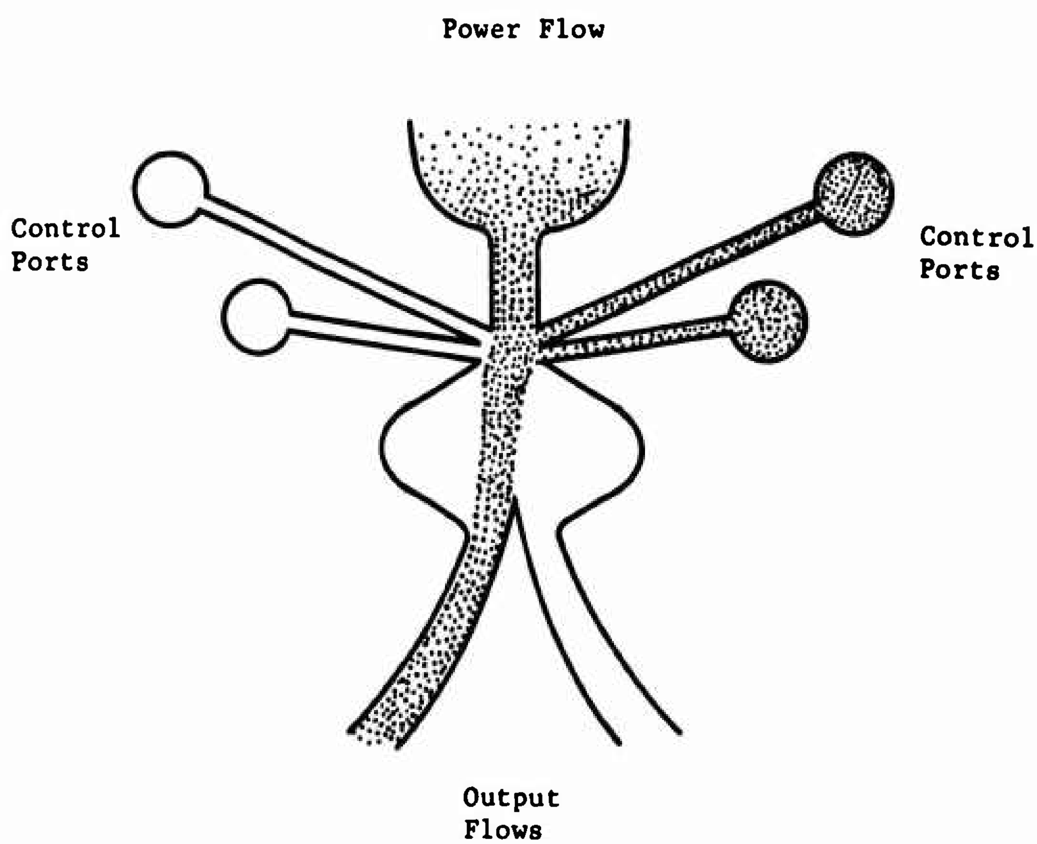

Figure 100. Summing Fluid Amplifier.

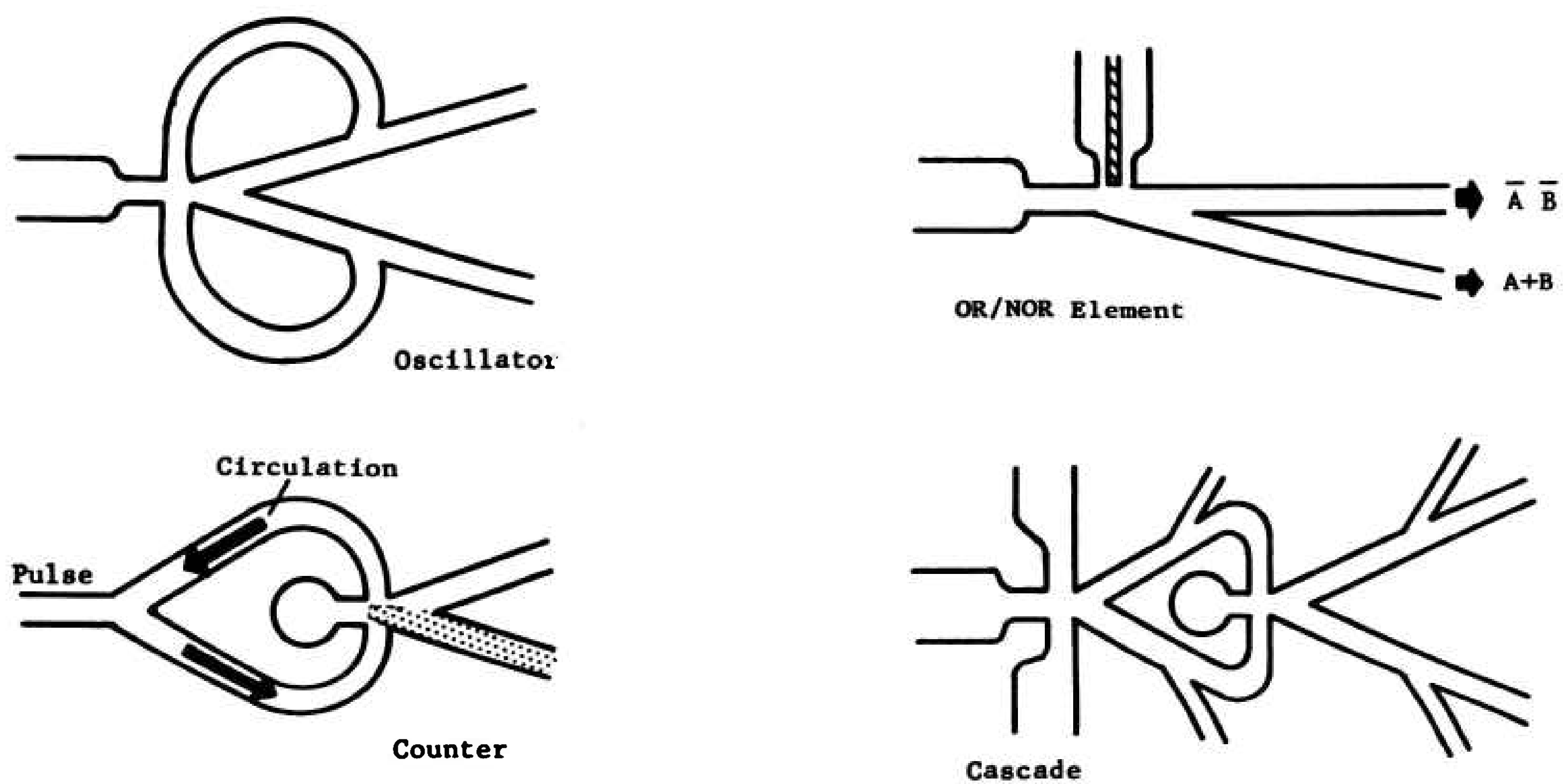

Figure 101. Bistable Amplifier Applications.

PROPORTIONAL AMPLIFIERS

Proportional amplifiers (Figure 99) are made by enlarging the interaction region and connecting the two side cheeks together with a low impedance path. The jet is then deflected by the momentum of the respective streams and issues from the output legs in proportion to the control streams.

SUMMING AMPLIFIERS

Summing amplifiers, needed in a feedback control system to compare references and actual signals, are made by adding a second set of control ports to a proportional amplifier (see Figure 100). The output is a function of the difference between the control signals.

VARIATIONS OF BASIC ELEMENTS

Computer elements, such as frequency oscillators, counters, OR/NOR elements, etc., are made from variations of basic elements (Figure 102).

The output gain of elements is measured by the ratio of output to control input. Gains are measured by the ratios of flows or pressures. The product of pressure and flow gain is termed power gain. Increased gain is obtained by cascading elements as shown in Figure 101. The extra set of outlets seen on bistable elements are vents. Vents allow elements to be cascaded without changing their characteristics so much by loading.

MANUFACTURE OF FLUID ELEMENTS

Fluid components are made of plastics, metals or ceramics and are machined, molded, etched or electroformed into the appropriate configurations. The materials employed must withstand the temperatures they are subjected to and be dimensionally stable and noncorrosive and resistant to erosion from the operating fluids. They must also be easily fabricated and scaled. Their configuration must allow them to be easily interconnected without mechanical connections which are easy to break or prone to leak. Integrated circuits fabricated in one piece without mechanical piping or couplers in between are preferred. Circuits are similar to an electronic etched circuit.

TWO-FLUID VORTEX CHAMBER FOR FUEL METERING

Under contract to AFSC, Honeywell Research (S&RD) has just completed a preliminary analytical and experimental investigation of two-fluid vortex chambers. Five experimental models as pictured in Figure 102 were built and tested. The fundamental characteristic is the establishment of a stable cylindrical interface between two immiscible fluids of different density. Among possible applications is included possible employment as an interface device to control a liquid pressure or flow rate with a

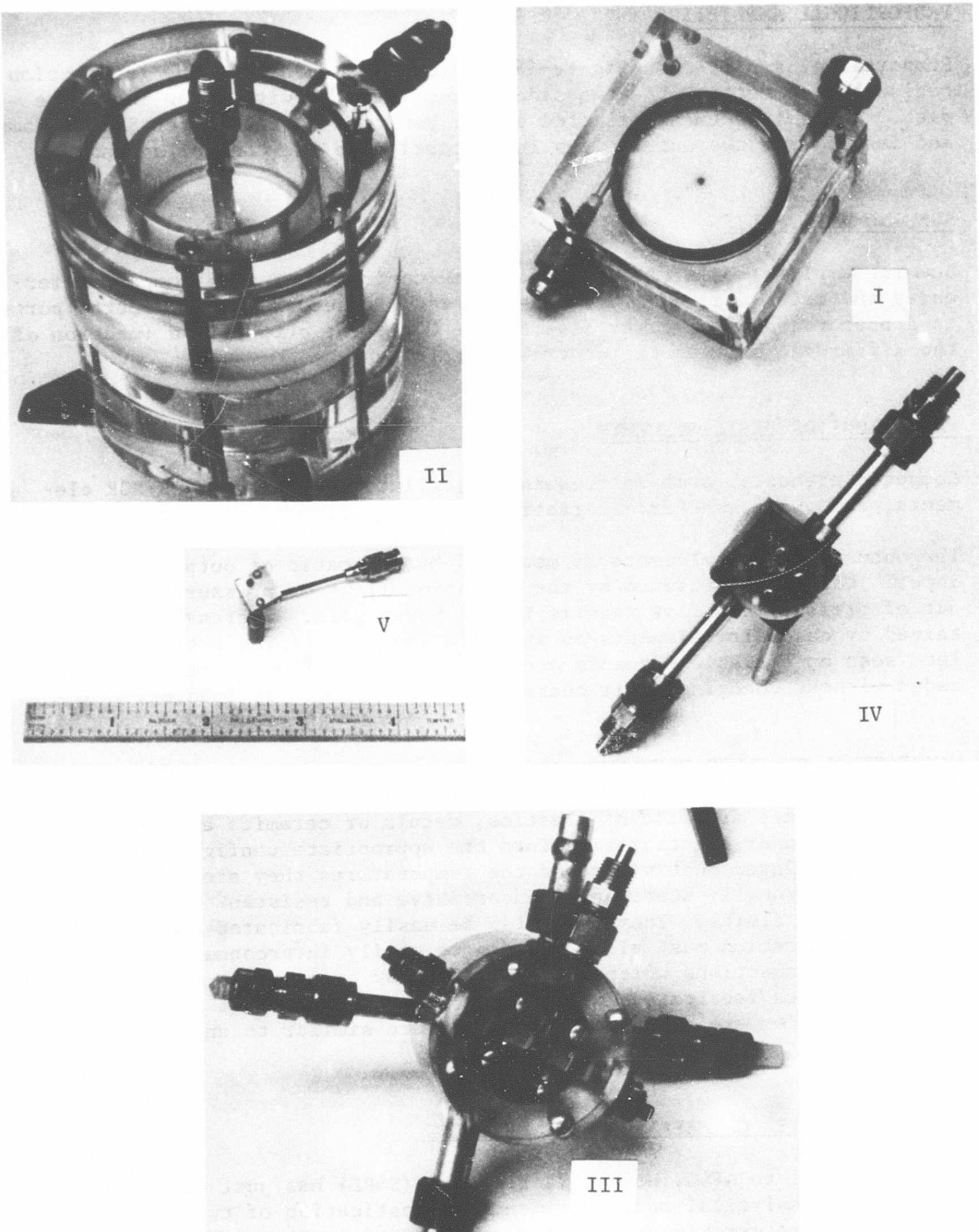

Figure 102. Photograph of Experimental Model Two-Fluid Vortex Chambers.

pneumatic signal input.

DESCRIPTION

The geometry of the chamber is shown schematically in Figure 103. Liquid enters the vortex chamber tangentially and exits through a circular slit in the floor at the periphery of the chamber. A gas is introduced at the center of the chamber. The centrifugal vortex flow causes the low density gas to move to the center. The high density liquid is whirled to the outside of the chamber.

The gas liquid interface is maintained at a radius inside the exit annulus. There is no flow of air, and hence, the fluids do not mix. The centrifugal forces which establish the interface can be made large so that the device can be made insensitive to gravity. A picture of the interface is shown in Figure 104.

At the interface surface, the gas pressure and liquid pressure are equal. Due to the swirling motion of the liquid, there is a centrifugal pressure gradient in the radial direction so that the liquid pressure increases radially outward from the interface. If the gas pressure is increased, the radius of the interface will increase until the balance of liquid and gas pressure at the interface is reestablished. Varying the interface radius with the gas pressure has no effect on liquid flow rate through the device. The liquid pressure distribution in the chamber depends on the liquid inlet supply pressure and the liquid back pressure on the exit annulus. Thus, the radius of the interface is a function of three pressures: the gas pressure, the liquid inlet supply pressure and the liquid back pressure.

When the interface is at equilibrium, there is no radial flow of liquid inside the exit annulus. If one neglects shear stresses at the end walls of the chamber, the liquid obtains the shear free solid body profile

$$\omega = \frac{V}{R} = \text{Constant}$$

as illustrated in B, Figure 103. This simplifying assumption yields the equations

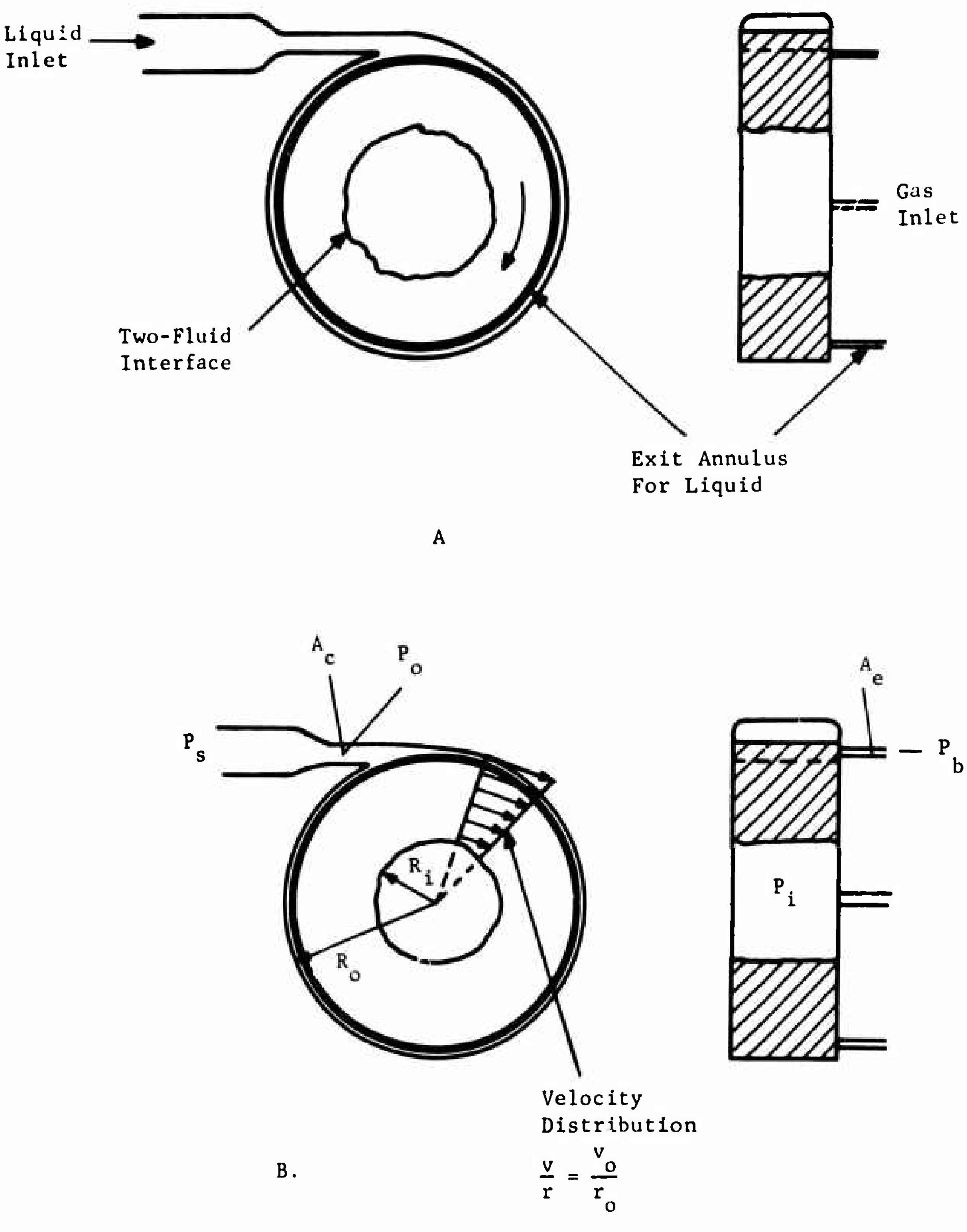

Figure 103. Functional Diagram of Two-Fluid Vortex Chamber.

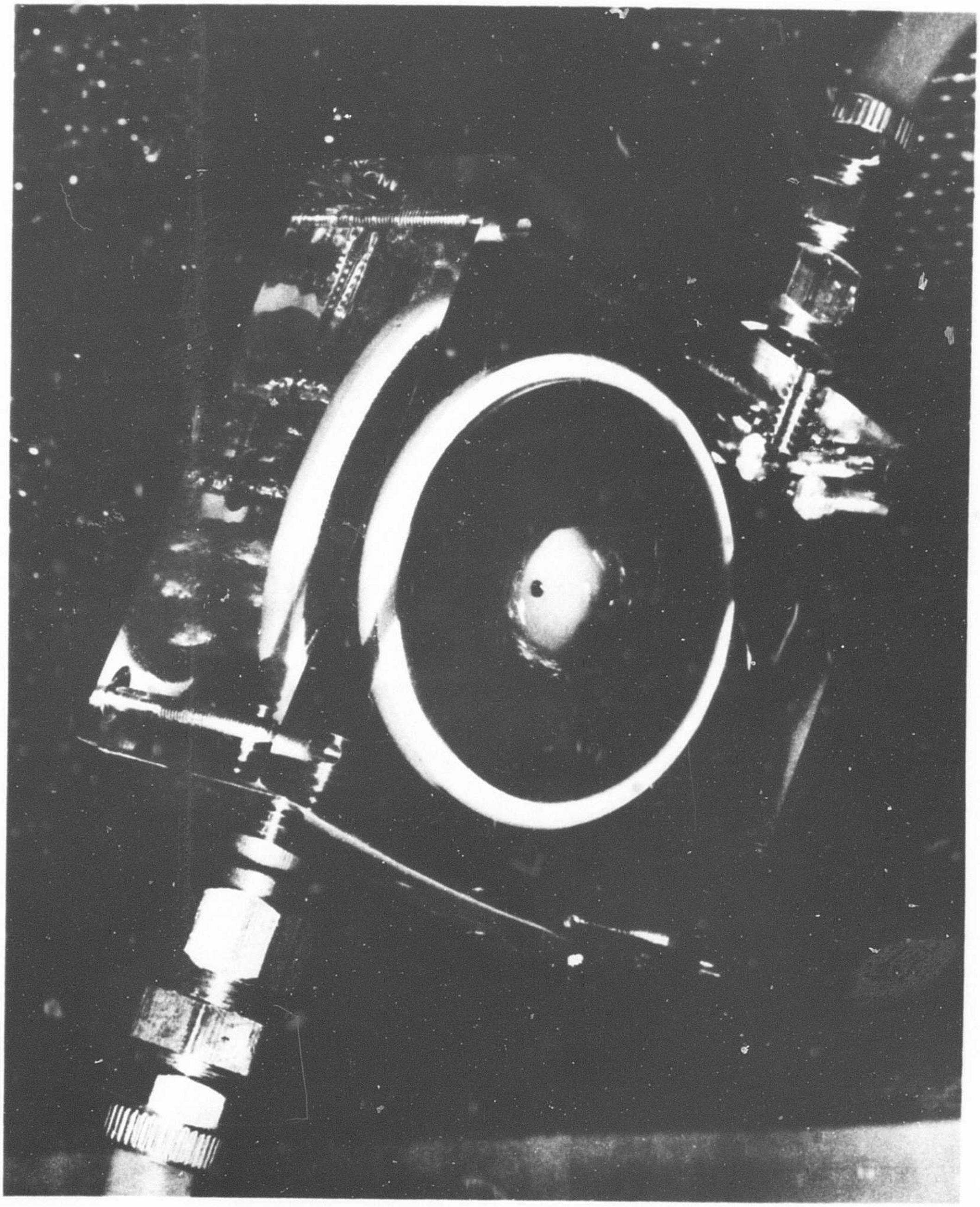

Figure 104. Photograph of Operational Two-Fluid Vortex Value.

$$P_o - P_b = \frac{1}{2} \rho \frac{Q^2}{A_e^2}$$

$$P_i - P_b = (P_s - P_b) \frac{\left[\frac{1}{A_e^2} + \frac{1}{A_c^2} \frac{R_i^2}{R_o^2} - 1\right]}{\left[\frac{1}{A_c^2} + \frac{1}{A_e^2}\right]}$$

where

Q = Liquid flow rate

P_o = Pressure at radius R_o

P_i = Pressure of gas and liquid at the interface

P_b = Liquid back pressure

P_s = Liquid supply pressure

A_e = Area of exit annulus

A_c = Area of the tangential inlet

R_i = Radius of the interface

R_o = Radius of chamber and exit annulus

The equation for P_i-P_b shows that the radius of the interface increases if the gas pressure, P_i, is increased and decreases if the liquid supply pressure, P_s, is increased.

Further, to maintain the interface at a constant radius there must exist a proportional variation of the gas pressure and the liquid pressure.

FUEL METERING

In the engine application, it is desired to have a direct relationship between a pneumatic gage pressure and the fuel flow to the engine. The fuel flow to the engine is a function of the nozzle area, the fuel manifold pressure P_s, and the compressor discharge pressure P_d which is the back pressure on the nozzles. Assume the following conditions: the nozzle area is fixed or depends on the flow rate, a hydraulic probe is used to adjust P_s so as to maintain the interface at a constant radius,

and the ratio of the exit area to the tangential inlet area is small. Then R_1 is constant and

$$Q_f = f\ (P_s - P_d) = f\ K\ (P_c - P_a)$$

where

Q_f = Fuel flow

P_s = Fuel manifold pressure

P_d = Compressor discharge pressure

P_c = Pneumatic control pressure

P_a = Ambient pressure

A method of mechanizing a pure fluid metering valve is illustrated in Figures 105 and 106. The probe shown in Figure 105 has been used to indicate, in an on-off fashion, whether the interface is inside or outside of a given radius. The probe consists of a fuel jet issuing from one end wall and received at a port on the opposite wall. When the interface is outside the probe radius the jet is free and undeflected. In this condition a high pressure is recovered at the receiver port. When the interface is inside the probe radius the jet is submerged and deflected, yielding a low pressure at the receiver port.

In Figure 106, the liquid inlet supply to the vortex chamber is proportional to the pressure output, P_n, from the fluid amplifier. If changing fuel pump pressure causes P_n to increase, flow input to the vortex chamber increases. The resulting increase in P_s over the set point pressure P_s* causes the interface radius R_i to decrease to less than the set point radius R.* The probe jet is deflected and a low pressure is recovered at the probe receiver port. This in turn causes the biased, conventional amplifier to bypass more flow and thus to lower the nozzle pressure P_n.

If P_s falls below P_s* the interface is outside the probe radius, and the control signal at "B" causes the amplifier to deflect more flow into the nozzle load and raise P_s. The system thus operates in a bang-bang fashion to regulate the fuel pressure at the set pressure P_s.*

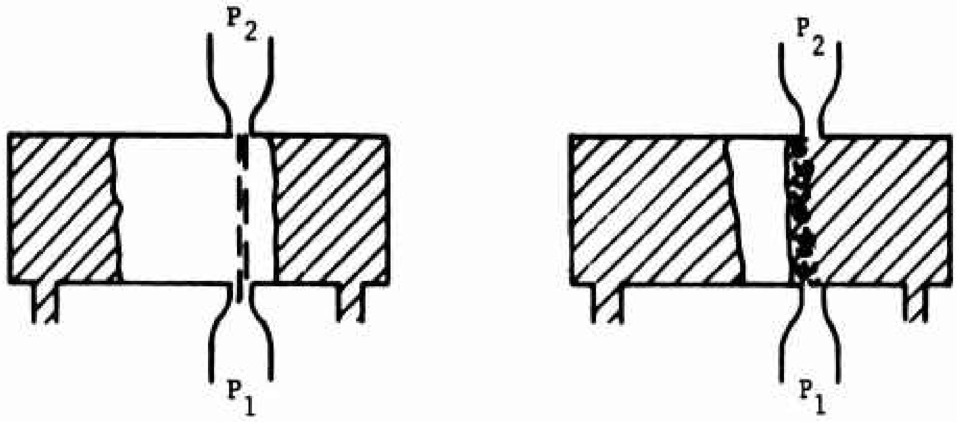

Figure 105. Schematic Diagram of Interface Probe.

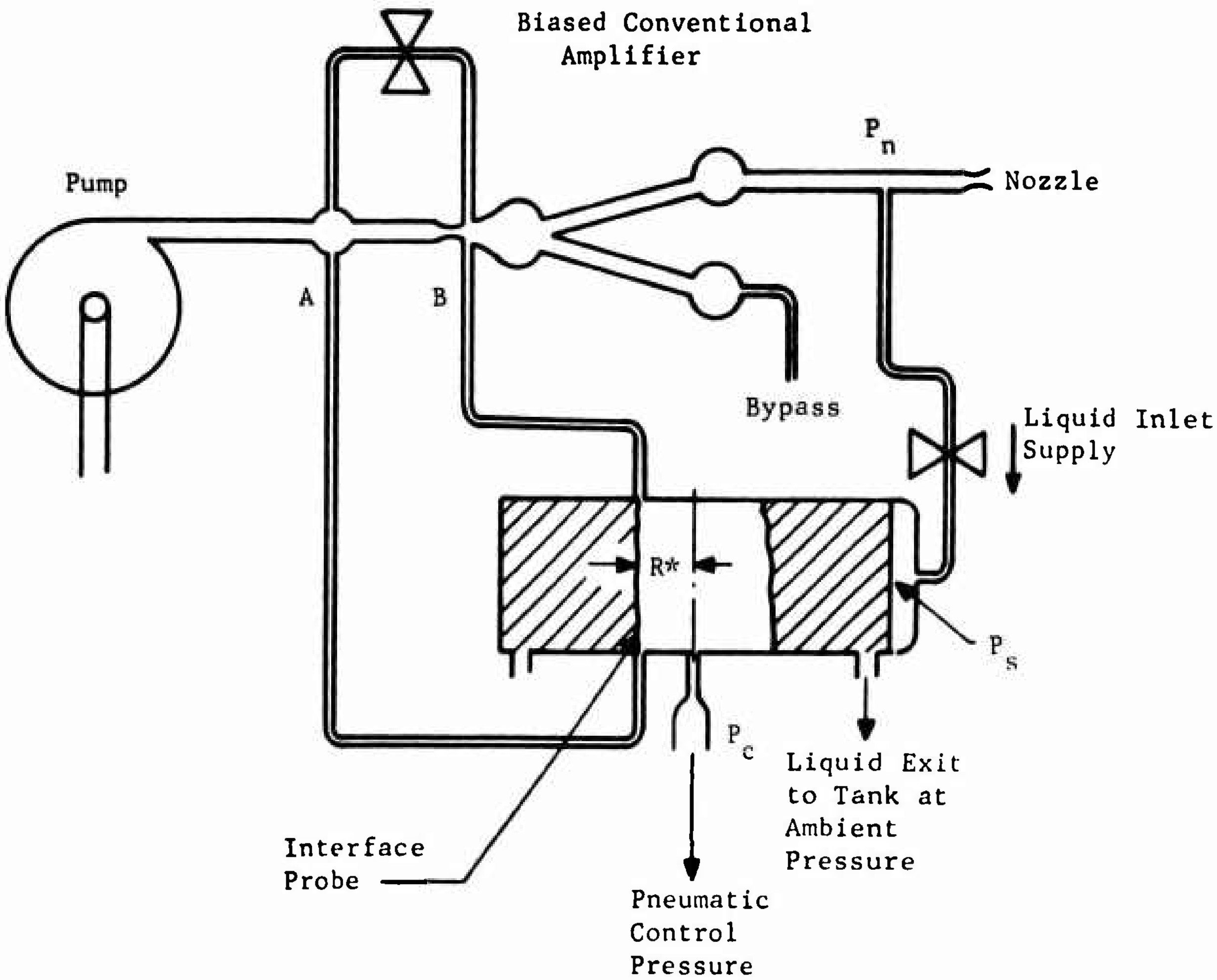

Figure 106. Schematic Diagram of Pure Fluid Metering Valve.

If the gas pressure is varied by a pneumatic control system, the equation for P_1-P_b shows that the set pressure P_s* varies proportionally. The two-fluid vortex chamber can be used in this way as an interface device to control fuel flow rate with a pneumatic control signal.

AIR ENTRAINMENT

A refinement is needed to prevent air from being entrained in the fuel stream to the nozzle when the air/fuel interface radius exceeds the radius of the probe location. This device should permit fuel in legs A and B (See Figure 106) to flow only toward the vortex chamber. This requirement has been met in the laboratory by a special arrangement of circuitry associated with a bistable amplifier, as illustrated in Figure 107.

In Figure 107, pressurized fuel is applied to the power port of the fluid amplifier and is also metered through two orifices into the control ports and into two vortex chamber probes. One probe is located on the centerline, and the other is located at the desired control radius R.* The orifice nearest the C_1 port has the greater restriction and biases the amplifier to output port O_1. However, if the radius of the air/fuel interface decreases, the fuel covers the opening of the left-hand probe and restricts the effluent at this probe, causing a back pressure buildup at C_1. Increasing P_c switches the output to O_2. The O_2 output is employed in the circuit (reference Figure 106) to accomplish the correction necessary to move the interface radius R_1 back to the set radius R.*

NEED FOR FURTHER WORK

Principal disadvantages of the vortex metering valve are (1) the need to run the liquid exit lines back to a fuel tank which is vented to ambient pressure, and (2) a relatively poor frequency response. To check frequency response, the gas pressure was varied with a pneumatic wave generator having both square and sine wave outputs. The response to the square wave input showed a 25-millisecond rise time. Frequency response data indicate a second-order system with a pure transport delay of 21 milliseconds. The dynamic data indicate a much poorer response than was predicted by a model in which the swirl energy was neglected. An improved model to account for the swirl has been formulated but is not yet completed. The frequency response can probably be improved by using a smaller diameter vortex chamber.

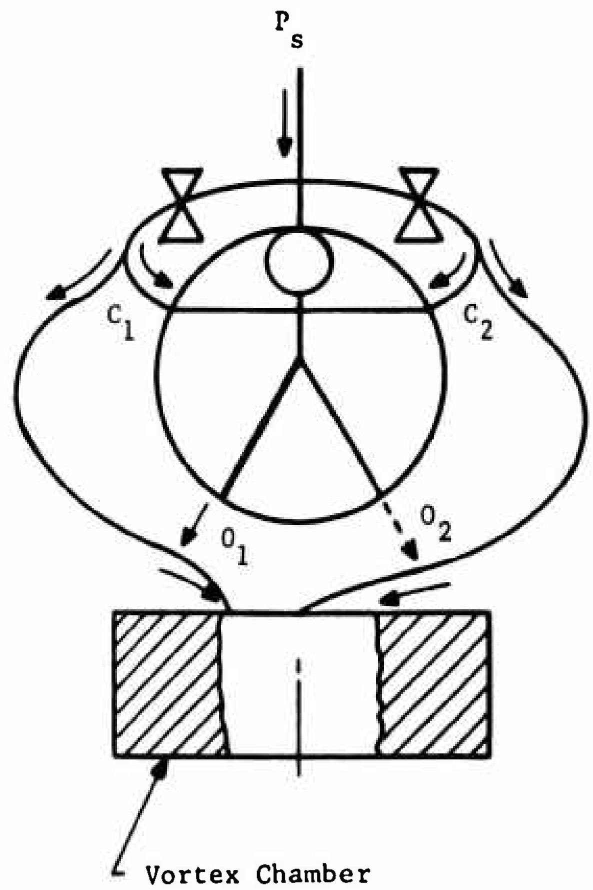

Figure 107. Back Pressure Control Circuit.

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The report contains conclusions and recommendations relative to obtaining an advanced generation of controls and accessories for small engines. Particular attention is given to improved packaging of the controls and accessories with emphasis on decreasing weight and volume and improvement in maintainability.

Engine starting is investigated in detail and it is concluded that a hand charged hydraulic accumulator/hydraulic motor-pump arrangement best meets the requirements of minimum weight, self-sufficiency and austere site operation.

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